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RESIDENTIAL OIL FURNACE SYSTEM OPTIMIZATION

PHASE II

by

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ABSTRACT

The report describes the second phase of an investigation into ways to improve the air pollutant emission and thermal efficiency characteristics of residential oil furnaces. A prototype, low-emission, warm-air furnace, designed in the first phase to embody a number of burner and combustor criteria for minimizing emissions compatible with high efficiency, was assembled and tested. Design details were changed as necessary during laboratory testing to help achieve the objectives. Applicability of the design criteria within current conventional oil-heat industry practices was demonstrated. Compared with estimated average characteristics of existing installed residential furnaces and boilers, NO_x emissions were reduced by 65% or more, and steady-state efficiency was increased by a minimum of 10 percentage points. Experimental results and component changes made in obtaining them were incorporated into a preliminary design for an integrated low-emission furnace which should be commercially producible and cost-competitive.

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SECTION I

CONCLUSIONS

1. Residential space heating systems can be designed in ways that reduce NO_x emissions substantially and that are also compatible with: (1) low emissions of carbonaceous air pollutants, (2) high thermal efficiencies, and (3) conventional oil heating industry practices in manufacturing, marketing, installing, and servicing units.

2. Design criteria for low emission oil burners and combustion chambers, derived from earlier research, have promising potential in the development of efficient low-emission warm-air furnaces and hydronic boilers.

3. Although appreciable reductions in NO_x emissions may be obtained by partial application of the design criteria (e.g., by retrofitting low-emission components into existing furnaces or boilers), maximum benefits can be achieved only by considering design optimization of the entire heating unit. By using the latter approach, supplementary concepts which improve efficiency or otherwise reduce residential fuel consumption also can be integrated into the total design.

4. The prototype, low-emission, warm-air furnace, built and tested to demonstrate proof-of-concept, showed the validity of these conclusions by achieving: 65 to 70% reductions in NO_x emission levels; acceptably low CO, UHC, and smoke emission levels; and steady-state efficiencies approaching the 85% maximum for noncondensing flue gas systems.

5. Cycle-averaged efficiencies, which are more difficult to measure and less well documented for existing residential heating equipment, were estimated to be a minimum of 10 percentage points higher with the prototype furnace than the average of central residential

oil-fueled furnaces. High steady-state efficiency contributed only part of that increase; devices designed to reduce standby heat losses (e.g., a draft damper in the combustion air supply and a sealed air system) accounted for the rest.

SECTION II

RECOMMENDATIONS

The full value of the low-emission technology developed in this investigation may be realized by applying it commercially. To further investigate and demonstrate potential benefits of commercialization, it is recommended that plans now being made to perform field testing of several optimum low-emission furnaces in actual residences be carried out.

The planned program will be conducted in two phases. In the first phase, further refinements of the optimum furnace design will be effected, partially to further optimize emissions and efficiency performance and partially to improve commercial producibility. Included will be analytical and experimental investigations for: (1) simplifying fabrication and reducing the mass of the firebox, (2) improving heat exchanger effectivity, (3) ensuring adequate performance at low ambient temperatures, and (4) satisfying applicable codes and standards. In the second phase, the finalized design will be used to construct approximately six low-emission furnaces for residential field testing during the 1977-1978 heating season. Operation in different types of residences and in at least two different climates is expected to yield definitive data on the achievable levels and constancy with time of air pollutant emissions and steady-state efficiency, as well as cyclical and season-averaged efficiencies, general operability, and any unusual service requirements.

SECTION III

INTRODUCTION

This report documents the second phase of a research program to establish technology for optimization of residential oil heating systems with respect to minimizing pollutant emissions and increasing heating system thermal efficiency. General overall goals have been to reduce emissions of oxides of nitrogen to less than 0.5 g NO/kg fuel burned, while maintaining minimum emissions of CO, UHC, and smoke, and to increase overall season-averaged furnace energy efficiencies by 10% or more above those achieved by current conventional systems. Emphasis was also placed on minimizing departures from existing heating industry manufacturing, distribution, installation, operation, and servicing practices which would be required to implement the developed technology.

The current research program is one of a series which Rocketdyne has carried out for the Environmental Protection Agency. The series began with an intensive investigation of residential and commercial oil burners (Ref. 1) which led to criteria for optimizing conventional burner designs with respect to pollutant emissions. For high-pressure atomizing, luminous-flame burners, it was found that:

1. Uniform mixing is beneficial. At a given overall burner stoichiometric ratio, NO_x production is reduced by minimizing local deviations from that overall ratio.
2. High-temperature adiabatic eddies embedded in the flame zone should be avoided. Long gas residence times in such eddies increase the production of NO_x.
3. Eddy recirculation near cool surfaces, on the other hand, may be beneficial. They help to reduce NO formation by supplying partially cooled vitiated combustion gases to the flame zone and, by dilution, lower flame temperatures appreciably.

4. Uniform one-dimensional (plug) flow is preferable to strongly swirling flow. Strong swirl promotes higher NO_x production unless: (1) local stoichiometric ratios generally exceed 1.5 (i.e., greater than 50% excess air), (2) the foregoing cooled recirculation is induced, or (3) combustion is completed so rapidly that the combustor can be made very short (i.e., residence times at flame temperature are minimized).

Based on these criteria, minimum pollutant emissions were obtained with burners having: (1) no flame-retention device, (2) choke diameter related quantitatively to the firing rate, and (3) oversized internal peripheral swirler vanes which promoted reactant mixing but without creating excessive turbulent recirculation. These burner design attributes were all concerned with the burner "head", i.e., that portion of the burner which admits prepared reactants into the combustion chamber. For that reason, this development was referred to as the "optimum head."

In addition to minimizing formation of oxides of nitrogen, the optimum head could be fired in the laboratory with considerably less excess combustion air, without producing unacceptable levels of carbonaceous pollutant emissions, than is conventional practice. Reducing excess air decreases the sensible heat lost with the flue gases, so application of the optimum head also has a potential for increasing overall furnace fuel utilization efficiency.

A further aspect of the research reported in Ref. 1 was that some variations in the combustion chamber construction (chamber diameter and relative orientations of the axes of the burner and chamber) affected emission levels. The results clearly demonstrated that pollutant emissions are sensitive to the design of each component comprising a residential heating combustion system and to design interactions among the components. It became obvious, therefore, that *minimum* emissions could be achieved only by systematically optimizing the burner in conjunction with the combustion chamber as well as with the furnace operating mode.

Nonetheless, it was recognized that the optimum burner head alone, as a retrofit device for existing burners in existing furnaces, might be commercialized more rapidly and less expensively than could an entire optimized furnace. Thus, in addition to the current program's investigations toward delineating requirements for optimizing the entire furnace, a parallel study addressed the feasibility of direct commercialization of the optimum head (Ref. 2). Two newly manufactured warm-air oil furnaces were retrofitted with optimum heads made to simulate those which might be produced by commercial stamping and bending of stainless-steel sheet. From the test results, it was estimated that widespread retrofitting of old existing residential units could yield an average increase of about 5% in the season-averaged thermal efficiencies, and average reduction of about 20% in the NO_x emissions for those units retrofitted. Those estimated achievable improvements are both less than half the target gains of the current research program.

Phase 1 of the residential oil furnace system optimization studies provided an essential background to the studies delineated in this report. It has been documented in Ref. 3 and is summarized in the following subsection.

SUMMARY OF PHASE I

The first phase of the research program comprised four distinct tasks:

1. Systems Analysis, in which current designs and practices employed in residential heating were reviewed and analyzed to identify potential areas of improvement

2. Conventional Burner/Combustor Matching Experiments, in which the 1.0 ml/s (gph) optimum burner was tested in research combustors having a variety of sizes, configurations, and constructions to broaden the design optimization

3. Recirculation Burner/Combustor Matching Experiments, in which 1.0 ml/s (gph) burners embodying forced recirculation of burned gases (to vitiate the burners' combustion air and lower flame temperatures) were similarly tested in a variety of research combustion chambers

4. Data Evaluation and Systems Analysis, in which the results of prior tasks were synthesized to support preliminary conceptual designs for two prototype, low-emission, improved-efficiency residential heating units.

The first systems analysis task was concerned primarily with thermal efficiency. The average steady-state efficiency, based on the fuel's higher heating value, for all existing installed units is probably between 72 and 75%, while mean season-averaged overall efficiencies probably are between 60 and 65%. Heat convected up the flue accounts for over 90% of the inefficiencies. Current residential heating technology is based on flue gas temperatures being high enough to ensure an adequate draft in a furnace's firebox and to prevent moisture from condensing in (and corroding) the furnace or flue. This concept limits the maximum achievable steady-state efficiency to about 85%; the minimum 15% decrement comprises: approximately 6 to 7% latent heat of combustion generated moisture, 7 to 8% sensible heat of the flue gases, and 1/2 to 1% cabinet or casing conduction losses. In practice, the decrement usually exceeds 15% because flue gas temperatures exceed the minimum to prevent condensation, because excess combustion air is not minimized and, particularly for hydronic boilers, because casing losses become greater than the minimum 1/2 to 1% range.

Heat losses are greater during cyclical furnace operation than during steady state because heat continues to be conducted through the cabinet and convected up the flue during standby periods when the burner is not being fired. Cyclical casing losses for warm-air furnaces may be twice their steady-state magnitudes. For hydronic boilers, because most boiler components are at nearly the same temperature during standby as during firing, cyclical casing losses may be three or more times those during steady-state operation. Nonetheless, convection of heat up the flue during standby usually accounds for most of the decrement between steady-state and cycle-averaged efficiencies. When the burner is turned off, a natural draft flow of air continues to pass through the burner, into the firebox, etc., and up the flue. That draft air flow cools furnace components between firings and can reduce cycle-averaged efficiencies by as much as 15%, although the average is probably around 8 to 10%. Thus, season-averaged efficiencies are estimated to be about 10 to 15% lower, on the average, than steady-state efficiencies.

Season-averaged thermal efficiencies of oil-fueled space heating equipment can be increased by: (1) lowering the quantity of excess air which dilutes the combustion product gases, (2) lowering the temperature of the gases admitted into the flue, (3) lowering or eliminating the draft air flow through the combustion equipment during standby, and (4) increasing cabinet insulation. Additionally, fuel consumption can be decreased significantly if: (5) the burner firing rate is properly matched to the local design temperature and to the residence's thermal demand, and (6) outdoor air, rather than heated household air, is supplied to the burner and to the unit's barometric control device.
Item 4 was considered not to be an appropriate area for study in this program. Item 5 is not related to the design of the heat source, *per se*, but is related to how a heating unit fits in an overall residential heating system; therefore, this item was not studied either. In a similar vein, many design aspects of a residence and its heating system exert strong influences on thermal demand patterns and overall

fuel utilization efficiency, but neither influence directly the thermal efficiency of the heat source nor are controlled by its design. Thus, items 1, 2, 3, and 6 above were identified as major potential areas for improving warm-air furnace and hydronic boiler performance.

In the second task, an optimum low-emission burner was laboratory-tested at a fuel firing rate of 1.05 ml/s (1.00 gph)* in a variety of cylindrical combustion chambers having different diameters and lengths, burner orientations, and methods and degrees of wall cooling. It was found that, to reduce NO emissions to a target level of 0.5 g NO/kg fuel at low excess air levels (10 to 15%), the burner should be fired into a combustor having the following design attributes:

1. The walls should be cooled so that approximately 20% of the fuel's higher heating value is extracted from the flame zone. The combustor wall temperature should be as uniform as possible during burner firing, and an elevated wall temperature should be maintained during standby. These conditions were achieved better with 90 C (194 F) water as the combustor coolant than with warm air, but it was nearly possible to satisfy them with the latter fluid.
2. The inside diameter of the combustor should be 0.28 m (11 inches) or greater for a 1.05 ml/s (1.00 gph) firing rate. This parameter influences NO_x emissions strongly.

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*Throughout this report, burner firing rates consistently are stated to two significant figures to the right of the decimal point. They are controlled by the oil supply pressure and oil spray nozzle used. Nozzles are calibrated in gallons per hour and the conversion factor, 1.052(ml/s)/(gph), is used to calculate firing rates in SI units. For convenience, nominal ratings of burners are stated less precisely by neglecting the 5.2% difference in units, i.e., optimum burners are designated as being "1 ml/s (gph) burners" regardless of the oil nozzle used or actual firing level in a particular test series.

3. The effective combustion chamber length, from its end near the burner to the location where the furnace heat exchanger begins to quench the gas temperature rapidly, should be at least 0.5 m (20 inches) and perhaps as long as 0.75 m (30 inches). The shorter length is appropriate for minimum NO_x emissions, but the longer length may be required to avoid excessive carbonaceous pollutant emissions at low excess air levels.

4. The combustor may be either side-fired or tunnel-fired, whichever is convenient for a particular furnace or boiler design. Actually, lower NO_x emissions are produced by the tunnel-fired configuration, but the other criteria have been stated such that the more common side-fired configuration can meet the NO_x target level.

Combustion gas recirculation (CGR) and flue gas recirculation (FGR) burners also were assessed in research combustor experiments. The CGR burner extracted partially cooled gases from the combustion chamber and mixed them with the combustion air upstream of the burner's air fan. This burner was found to produce acceptably low NO emissions, but generally produced unacceptably high CO and UHC emissions. Only very limited sets of design and operating conditions were found where CO, smoke, NO, and operability were all acceptable but, even then, UHC concentrations remained high.

The FGR burner's combustion air was mixed with externally circulated flue gases obtained downstream of the furnace heat exchanger. Being cooler than combustion chamber gases, flue gas is a more effective flame-zone diluent. Steady-state, low-excess-air, operating conditions were found which had acceptably low emissions of all air pollutants but, when tested in cyclical operation, burner startup spikes of excessive emissions of carbonaceous pollutants were experienced. The amplitude of the spikes was lowered by reducing the amount of flue gas recirculated; however, NO production increased concurrently so that, when operation and carbonaceous emissions were acceptable, about 0.6 g NO/kg fuel was exhausted.

From the experimental results, it was concluded that the optimized conventional burner had better potential for minimizing emissions of air pollutants and maximizing efficiency than either the CGR or FGR burners. Therefore, it was incorporated in preliminary designs for candidate prototype, low-emission, residential heating units. One preliminary design was developed for each of the two common cooling media, namely, air and water. The warm-air furnace design was based on making appropriate modifications to an existing warm-air furnace of contemporary design. The hydronic boiler design, on the other hand, involved all new construction. EAch of these prototype design concepts was discussed with engineering personnel of a manufacturer of that type of residential heating equipment. Thereafter, the prototype warm-air furnace was selected to be built and tested in Phase II. The design is described in the next section.

SECTION IV

EXPERIMENTAL INVESTIGATION OF THE PROTOTYPE LOW-EMISSION FURNACE

An experimental, 1 ml/s (gph), prototype, low-emission, oil furnace was constructed and tested. The objectives were to construct a prototype system embodying the essential concepts and design features generated in Phase I and to evaluate its capabilities to satisfy the emission control and performance design goals. As reported in Ref. 3, a selection was made to construct a warm air prototype furnace as opposed to a hydronic type system.

The experimental prototype warm-air furnace was constructed using a stock commercially available unit as the primary component base, with modifications to provide the optimized burner and firebox components, as well as other appropriate design features.

The designs of nonstandard components were such that they were amenable to fabrication by conventional production techniques, even though those techniques were not used on this one-of-a-kind unit. The furnace modifications were designed such that further modifications could be incorporated as suggested by subsequent test results.

The general overall performance goals for the prototype furnace were:

1. To reduce air pollution emissions to or below the following levels:
    a. Oxides of nitrogen, 0.5 g NO_x (as NO)/kg fuel burned
    b. Carbon monoxide, 1.0 g CO/kg fuel
    c. Gaseous hydrocarbons, 0.1 g UHC/kg fuel
    d. Smoke, No. 1 on the Bacharach scale

In comparison with average emission levels reported from a field survey of actual residential heating units (Ref. 4), these goals sought not to exceed the average CO and UHC emissions from burners in their as-found conditions while reducing smoke and NO_x emissions by 68% and 72%, respectively, from their reported average levels.

2. To increase cycle-averaged thermal efficiency by 10% or more above the mean achieved by existing installed residential heating units
3. To comply with all applicable safety codes and operational standards
4. To the extent possible:
    a. To remain cost competitive with currently manufactured units
    b. To decrease operating noise
    c. To minimize unit volume

Emphasis was placed on developing advanced technology that can be implemented by the U.S. heating industry with minimum departures from current manufacturing, distrubution, operation, and servicing practices.

Design options were evaluated on the basis of a newly manufactured product line, with compromises involving retrofit versatility given a much lower priority. However, product saleability certainly was of concern and, therefore, unit costs and acceptability to manufacturers, servicemen, and customers were considered in the design selection process.

EXPERIMENTAL APPARATUS

Stock Furnace Prior to Modifications

As indicated above, the experimental, prototype, low-emission furnace was obtained by making appropriate modifications to a stock commercially available warm-air furnace. The unit selected to fill this role was a Lennox Model 011-140 furnace manufactured by Lennox Industries, Marshalltown, Iowa. Illustrated in Fig. 1, reproduced from a Lennox brochure, the Lennox 011 series is an outstanding example of contemporary residential oil furnace design. It is more compact than current models offered by most manufacturers, primarily because of its unique-design compact heat exchanger, and is capable of achieving quite high steady-state efficiencies. The main design features of the stock Lennox 011-140 upflow furnace are summarized in this subsection.

The stock furnace was equipped with a Lennox flame-retention-type burner having radial slots in its choke plate and a short convergent enclosure downstream of the choke plate. The recommended firing rate range is 0.89 to 1.05 ml/s (0.85 to 1.00 gph), and it is about the most compact unit in its heating capacity range, with a 1.45 m high by 0.66 m wide by 0.61 m deep (57 by 26 by 24 inches) cabinet. The firebox is a refractory-lined, side-fired-type enclosure. The firebox lining is a single piece of molded refractory fiber material, approximately 0.25 m (9.8 inches) inside diameter. It has a "corbel" top, with a 0.18 m (7.0 inches) exit diameter.

The heat exchanger section is approximately 0.33 m (13 inches) high, 0.61 m (24 inches) wide and 0.444 m (17.5 inches) deep. Its primary heat exchange section consists of an uninsulated 0.279 m (11 inches) diameter central steel cylinder (an extension of the outer shell of the firebox) with a rearward facing exit that channels the combustion gases into a rear manifold. From there, the gases are distributed among six, flat, heat exchanger panels with combustion gases inside moving toward

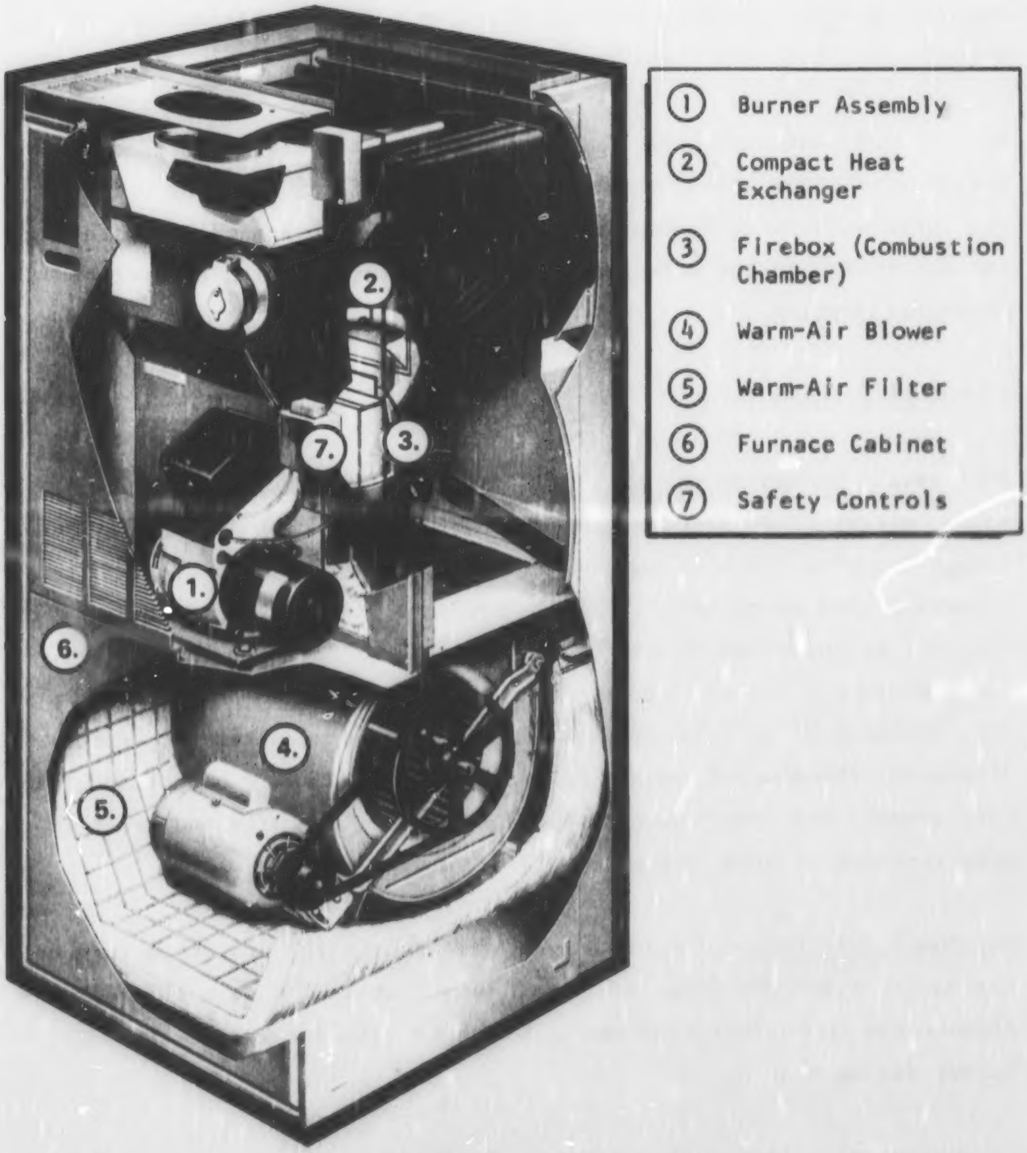
① Burner Assembly
② Compact Heat Exchanger
③ Firebox (Combustion Chamber)
④ Warm-Air Blower
⑤ Warm-Air Filter
⑥ Furnace Cabinet
⑦ Safety Controls
2.
7.
3.
1.
6.
4.
5.

Figure 1. Cutaway Drawing of the Stock Lennox Model 011-140 Furnace (Reproduced from a Lennox Industries Inc. brochure with their permission)

the front of the furnace, and coolant air outside flowing vertically upward. Combustion gases are collected by a front manifold that discharges them into the flue.

The firebox/heat exchanger enclosure utilizes the air-gap method of insulation on three sides, with the warm-air baffling standing off from the outer wall about 0.00137 m (0.5 inch). The common wall separating the vestibule and the warm-air channel is insulated with a layer of fiberglas matting.

Prototype Optimum Furnace

The overall design of the prototype optimum low-emission furnace is shown in Fig. 2, an assembly layout drawing used in building it. Those design features which differ substantially in concept from their counterparts in the stock Lennox furnace also are illustrated pictorially in Fig. 3. As described in the following paragraphs, major changes were made concerning the oil burner, the firebox, and the combustion air supply, while minor modifications were made in several other components. Otherwise, the stock furnace external cabinet, warm-air blower and filter, compact heat exchanger, and all electrical circuits and controls were retained without change.

Optimum Burner Unit - The oil burner utilized in the prototype furnace design was a Beckett Model AF burner body fitted with an optimized non-flame-retention burner head and a device to eliminate draft air heat losses during standby.

The optimized burner head consisted of six air swirl vanes, canted 25 degrees from the blast tube centerline, and a firing-rate-dependent choke diameter. The air swirl vanes were relatively large; they were approximately 0.05 m (2 inches) long, and extended from approximately 0.03 m (1.2 inches) diameter out to the diameter of the blast tube.

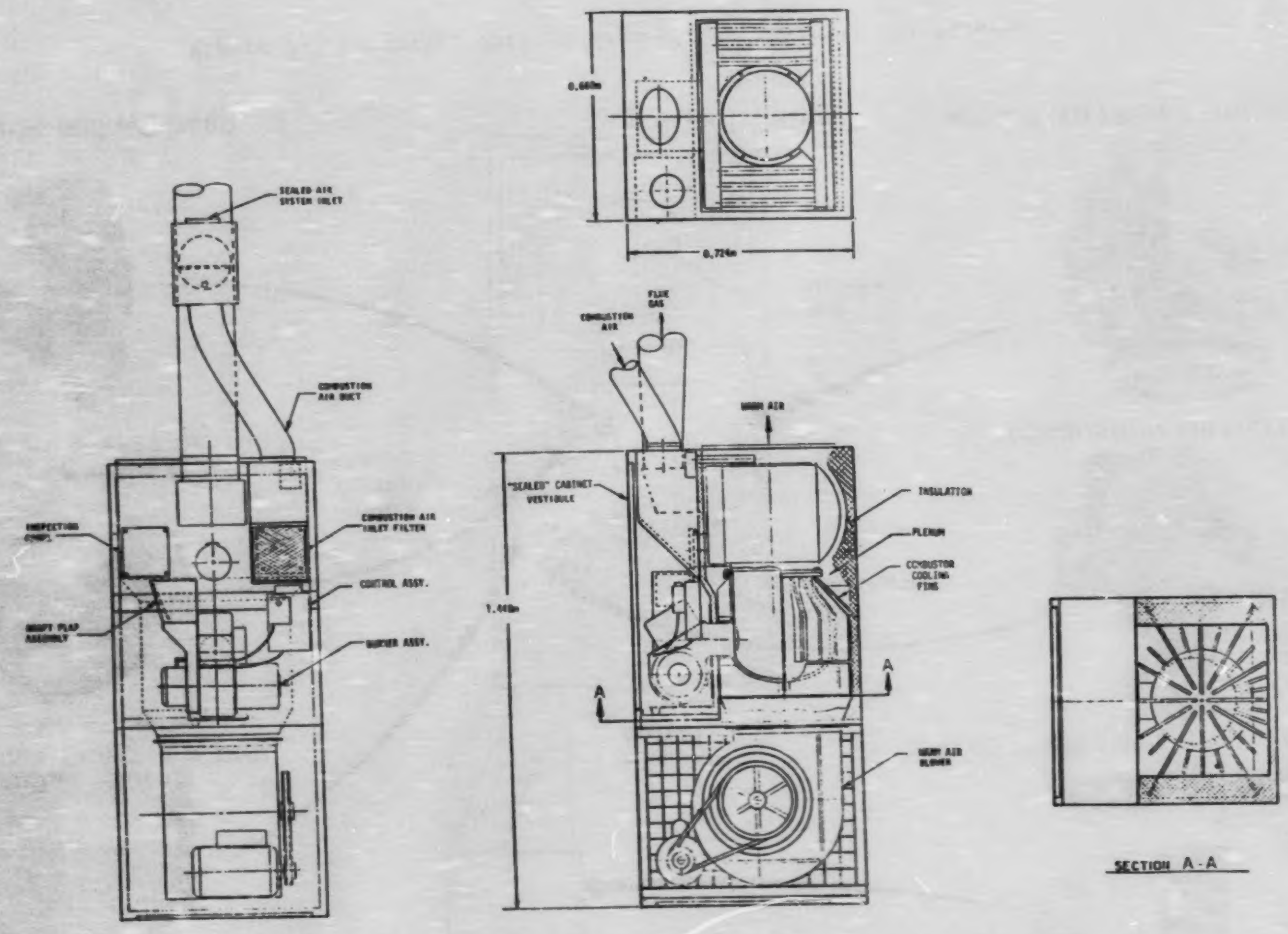
SEALED AIR SYSTEM INLET
COMBUSTION AIR DUCT
INSPECTION COMPT.
COMBUSTION AIR INLET FILTER
CONTROL ASSY.
DRAFT FLAP ASSEMBLY
BURNER ASSY.
0.660m
0.724m
COMBUSTION AIR
FLUE GAS
WARM AIR
"SEALED" CABINET VESTIBULE
1.448m
INSULATION
PLENUM
COMBUSTOR COOLING FINS
A
A
WARM AIR BLOWER
SECTION A-A

Figure 2. Layout Assembly Drawing, Prototype Optimum Warm-Air Oil Furnace

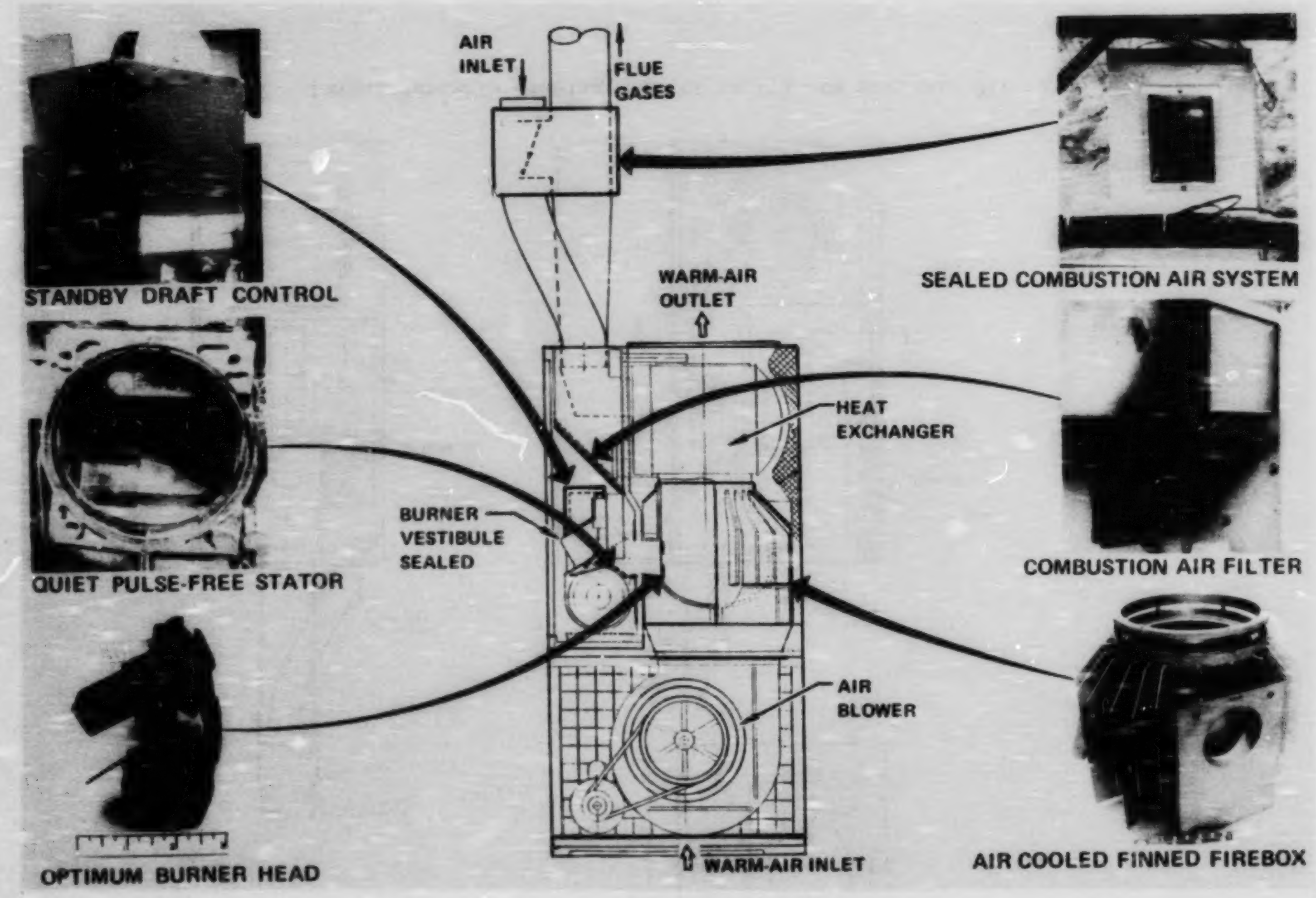
AIR INLET
FLUE GASES
SEALED COMBUSTION AIR SYSTEM
STANDBY DRAFT CONTROL
WARM-AIR OUTLET
HEAT EXCHANGER
QUIET PULSE-FREE STATOR
BURNER VESTIBULE SEALED
COMBUSTION AIR FILTER
AIR BLOWER
OPTIMUM BURNER HEAD
WARM-AIR INLET
AIR COOLED FINNED FIREBOX

Figure 3. Prototype Optimum Warm-Air Residential Oil Furnace

The choke diameter was related to the specific installed firing rate according to:

$$D = K_c \ (\dot{w}_{oil})^{0.4} \tag{1}$$

where

K_c = 0.037 for $\dot{w}_{oil}$ in ml/s and D in meters, or
= 1.488 for $\dot{w}_{oil}$ in gph and D in inches

The choke diameter for the 1 ml/s (gph) burner was 0.038 m (1.50 inches).*

The standby draft control device was an off-vertical flap valve in the air inlet to the burner's combustion air fan. The flap swung open under the suction effect of the operating burner fan and dropped to the closed position when electrical power to the burner drive motor was cut off.

After a large part of the testing had been conducted, some further modifications were made in the burner's combustion air passages. An 0.0826 m (3.25 inch) outside-diameter static disc was installed in the 0.0984 m (3.88 inch) inside-diameter blast tube to increase the pressure drop between the air fan and the combustion chamber and, thereby, to suppress any coupling effect between combustion perturbations and fan stall. The air discharge side of the squirrel-cage impeller cavity also was modified. This involved the addition of a "quiet, pulse-free stator," which is a skewed-lip, sheet-metal extension to the discharge lip of the impeller cavity, to reduce the impeller-to-casing gap from approximately 0.0064 m to 0.0016 m (Fig. 4). Thereby, the maximum output pressure of the fan was increased from 0.0318 m (1.25 inches) to 0.0381 m (1.50 inches) of water column. One reason that burner fan

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*This was erroneously described in Ref. 3 (page 58) as being 0.042 m (1.65 inches).

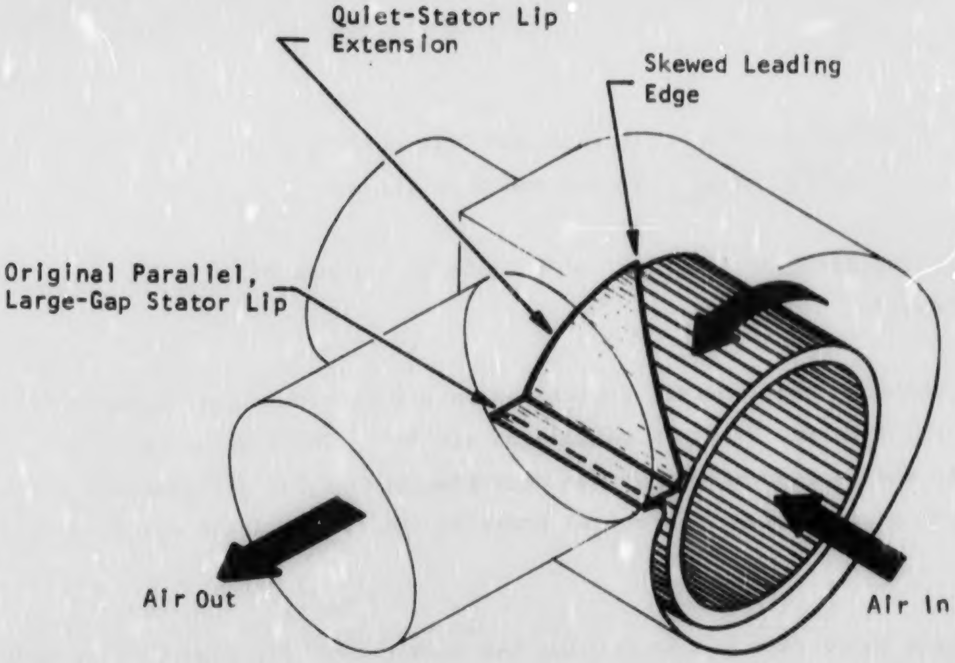
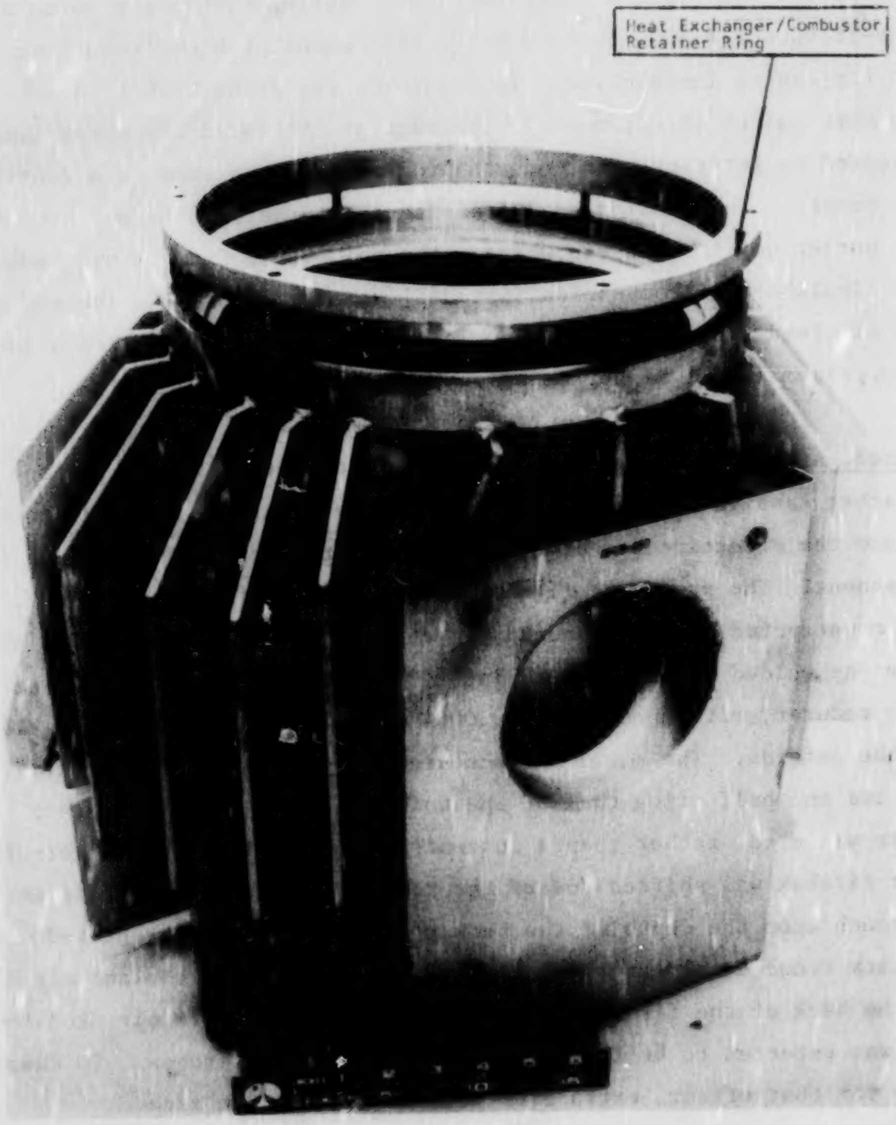

Quiet-Stator Lip Extension
Skewed Leading Edge
Original Parallel, Large-Gap Stator Lip
Air Out
Air In

Figure 4. Schematic of the Skewed-Lip "Quiet Stator" Extension for the Optimum Burner Combustion Air Fan

casings are made with a relatively large casing-to-fan gap is to avoid high-frequency (~500 to 1000 Hz) burner noise problems resulting from fan tip/casing interaction. To eliminate the production of discrete pressure pulses (i.e., noise), the edge of the added lip extension was oriented to interact with more than one blade at a time in a continuous manner. In addition to increasing the burner fan output and making the burner quieter (no tip whine), the pulse-free flow of air reduces the likelihood of combustion instability being externally induced by the air feed system, which would in turn result in combustion noise and off-optimum (i.e., higher emissions) combustion.

Finned, Air-Cooled Firebox - The firebox for the prototype furnace was a rather massive uninsulated steel assembly with external fins to increase the effectiveness of the external warm-air flow in cooling the component. The prototype firebox design is illustrated in Fig. 5. It was constructed from a standard Schedule 40, 12 inch (0.305 m) diameter pipe cap welded to a 12 to 10 inch (0.305 to 0.254 m) diameter eccentric reducer, with twenty-four 0.0064 m (0.25 inch) thick fins welded to the outside. The outer surface area was increased to approximately six and one half times that of the unfinned shell. The eccentric reducer was used, rather than a concentric one, so that the larger-diameter firebox was shifted toward the rear of the furnace and did not encroach upon the depth of the burner vestible. The rear-biased eccentricity reduced the external cross-sectional area for coolant air flow at the back of the firebox which, because the burner is directed toward it, was expected to be the hottest protion of the firebox. To compensate for that effect, extra fins were placed in that area.

The assembled prototype firebox mass was 57.2 kg (126 lbm), an increase of about 50 kg (110 lbm) over the refractory-fiber-lined stock firebox which is replaced. This massive construction was adopted intentionally to accomplish two objectives in addition to extracting heat from the flame zone; both are related to the heat sink nature of a massive combustion chamber. First, a massive heat-sink chamber can more readily


Heat Exchanger/Combustor Retainer Ring

5ZZ31-10/21/75-SID

Figure 5. Front View Photograph of the Finned, Air-Cooled Combustor for the Low Emission/High Efficiency Prototype Furnace System

approach uniform inside surface temperatures than can a lightweight, unlined, metal firebox. This is important for avoiding excessive NO formation and wall erosion (or possibly burnout) at "hot spots," and for avoiding smoke and UHC formation at "cold spots." The second objective is for the combustion chamber to retain most of its stored heat during standby periods so that the firebox is considerably warmer at burner startup than is the usual warm-air furnace practice. This objective arose from the observation that start-transient spikes in the emissions of CO, smoke, and UHC were considerably lower with water-cooled than with uninsulated, air-cooled combustors (Ref. 3).

The mating of the heat exchanger to the prototype firebox was accomplished by a bolted retainer ring compressing a pyroflex gasket between the finned firebox and the 17-gage heat exchanger shell. This type of firebox/heat exchanger attachment was selected for the experimental prototype furnace to avoid problems of metal fracturing due to differential thermal expansion between the two very different component thicknesses.

Combustion Air Supply - Two related modifications were concerned with the combustion air supply. The first was the use of a sealed air supply system and the second was the provision of a separate filter for the combustion air.

In the sealed air supply system, outdoor air is piped into the room where the furnace is situated, and is supplied both to the burner for combustion air and to the barometric control device for admixture with flue gases. The main advantage of doing this is that residential fuel consumption is reduced because heated (and perhaps humidified) air from within the residence is not consumed by the furnace, so that the residence's thermal demand is lowered. Experimental data reported in Ref. 5, from which the system was adapted, showed that sealed air systems can reduce fuel consumption by at least 5% and, in some installations, by as much as 15%. An insulated galvanized air duct conducts outdoor air to an air plenum constructed around the barometric control damper connection with the furnace's flue pipe.* The plenum serves as an air supply manifold where the air flow is divided, as required, between the barometric flue pressure control device and the oil burner.

As described in Ref. 5, the combustion air was ducted from the plenum directly to the burner inlet through a length of uninsulated flexible plastic tubing. The installation for the prototype furnace was different, in that the combustion air was ducted to a connection on the furnace, rather than directly to the burner. This was done to facilitate filtering the combustion air. A second air plenum was constructed within the furnace's burner vestibule, and formed the supply side of a flat, rectangular, fiberglas filter panel. The burner vestibule was also sealed so that the only source of combustion air was that which passed through the filter.

Combustion air filtration was considered to be especially important for the prototype optimum furnace because it was designed to operate with close to minimum (10 to 15%) excess combustion air. Accumulations of dust, lint, hair, etc., in the burner air passages may shift the operating point into a smoky and/or high CO condition more easily when the burner is tuned for low excess air operation than conventional burners operating with high excess air.

The combustion air filter was positioned so that, upon entering the burner vestibule, air passed directly over the furnace electrical controls, then completely across the vestibule to the burner's air entry. This arrangement was meant to promote cooling of the electrical controls, burner components, and furnace cabinetry, all of which were anticipated as having potential overheating problems because the combustion air flow was lower than the air flow through the stock furnace's vestibule before it was sealed. In addition to better component cooling, this vestibule flow pattern was also expected to help temper the outdoor air and at least partially offset a tendency for cold combustion air to increase emissions of carbonaceous pollutants.

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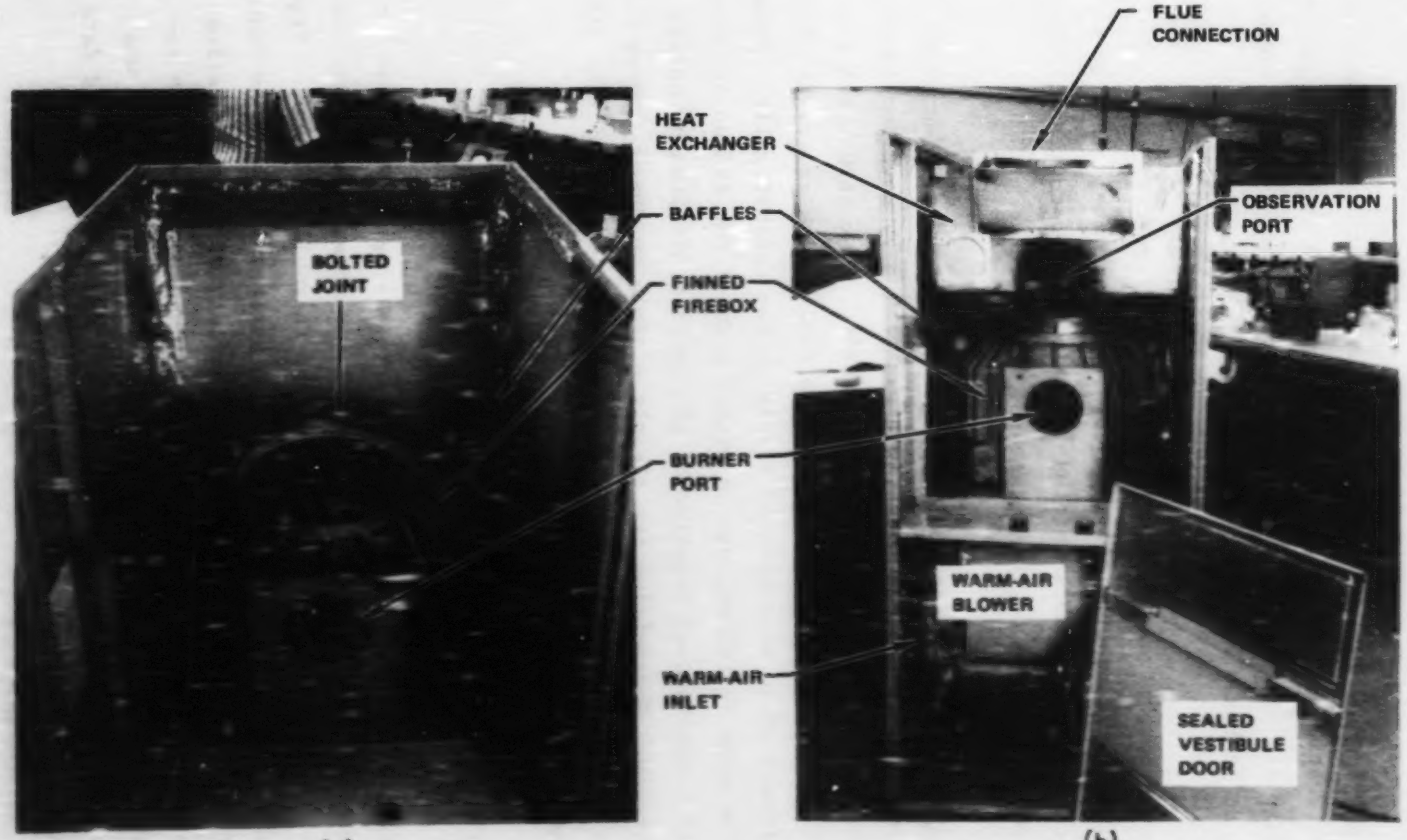
*The air supply duct was omitted from the installation of the prototype furnace in Rocketdyne's outdoor test laboratory. The sealed air plenum pictured in Fig. 3 was a prototype unit supplied by Lennox with the stock furnace.

The enclosed burner vestibule, together with the rest of the sealed air system, also should beneficially cut down on burner motor and fan noise, combustion noise, and odors emitted into the furnace room and, presumably, into other parts of the residence.

Other Modifications - Minor modifications were made to a few other furnace components. The burner vestibule was sealed by covering louvers and handholes in the vestibule closure panels. Some structural reinforcement was added inside the cabinet so that the extra weight of the finned firebox could be supported evenly by the cabinet walls. Finally, two baffles were installed, one on either side of the finned firebox, to prevent a substantial fraction of the coolant air from bypassing the finned firebox. The baffles are clearly visible in Fig. 6, which shows two views of the firebox in the prototype optimum furnace during its initial assembly.

Test Facility and Instrumentation

Performance of the prototype optimum furnace was evaluated in an outdoor laboratory facility having provisions for measurement of pollutant emissions, operational characteristics, and thermal efficiency. Figure 7 is a schematic of the furnace evaluation system; it shows the installation of gas and air flow ducting and a variety of instrumentation. Basic thermal performance measurement techniques conformed with requirements of ANSI Z91.1-1972 (Ref. 6). Other instrumentation was added to provide enlarged understanding of furnace behavior and data for calculating cycle-averaged thermal efficiency.


BOLTED JOINT
HEAT EXCHANGER
BAFFLES
FINNED FIREBOX
BURNER PORT
WARM-AIR INLET
FLUE CONNECTION
OBSERVATION PORT
WARM-AIR BLOWER
SEALED VESTIBULE DOOR

(a) (b)

Figure 6. Installation of the Finned Firebox and Compact Heat Exchanger in the Prototype Optimum Furnace

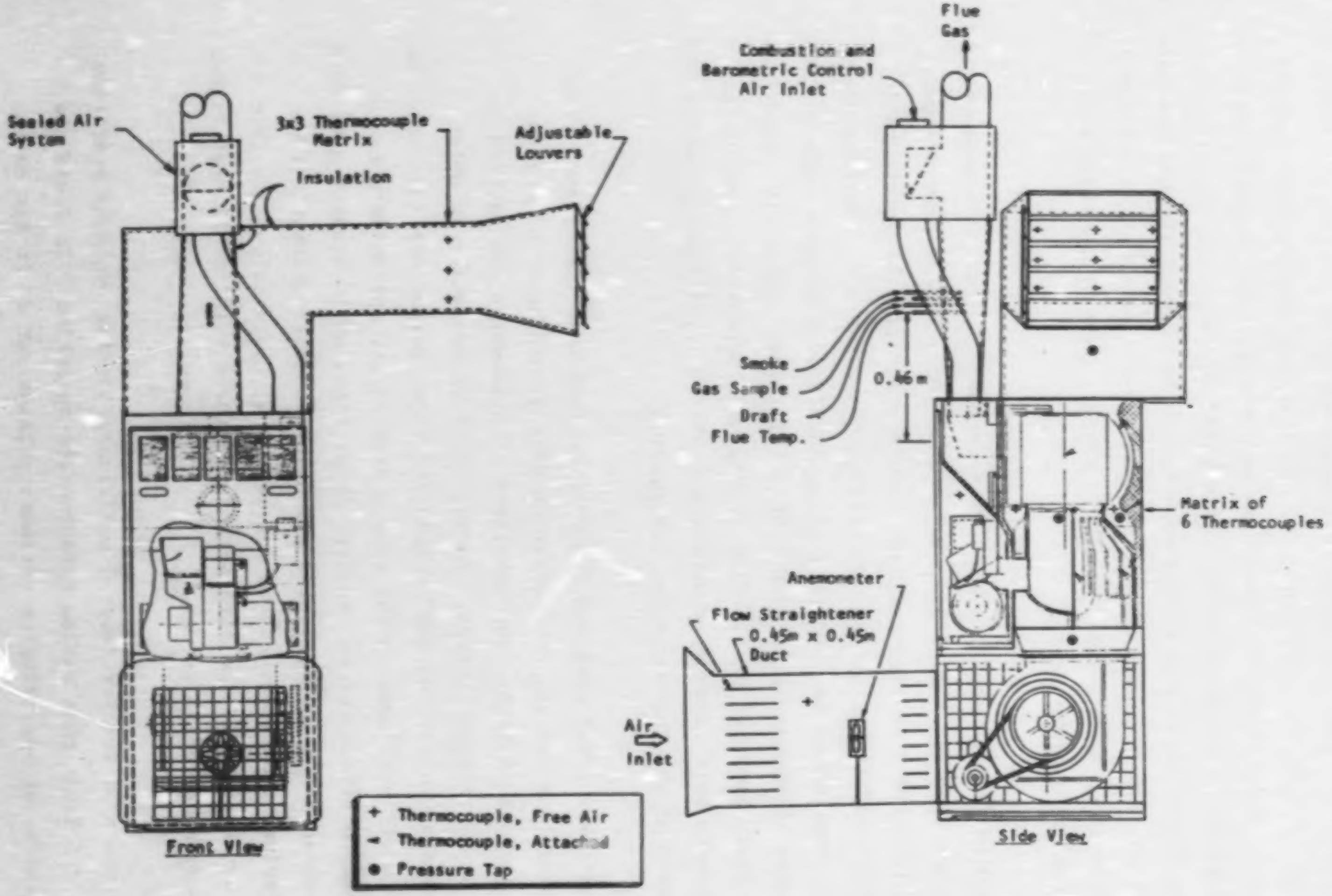

Sealed Air System
3x3 Thermocouple Matrix
Insulation
Adjustable Louvers
Front View
Combustion and Barometric Control Air Inlet
Flue Gas
Smoke
Gas Sample
Draft
Flue Temp.
0.46m
Matrix of 6 Thermocouples
Anemometer
Flow Straightener
0.45m x 0.45m Duct
Air Inlet
Side View
+ Thermocouple, Free Air
- Thermocouple, Attached
● Pressure Tap

Figure 7. Schematic of the Furnace Performance Evaluation System

Constituents in the flue gases were measured by continuously withdrawing a gas sample from the center of the flue, at the location denoted in Fig. 7, and passing it through an analysis train. The analytical system provided for continuous analyses for O_2, CO_2, CO, NO, and UHC species remaining in the dry gases after their passage through condensible traps, filters, and dryers. Details on the setup and operation of the train, instruments used and their ranges, data processing, etc., are given in Appendix A.

The furnace flue thermal losses were determined by making measurements to support flue gas heat balances. Combustion gas mass flowrate was back-calculated from measured fuel flowrate and stoichiometric ratio (as determined from flue gas composition measurments). The flue gas exhaust temperature was measured in an insulated flue pipe with an iron/constantan thermocouple located 0.46 m (18 inches) above the centerline of the heat exchanger exit. Flue draft, gas composition, and smoke measurements were taken at successive 0.0317 m (1.25 inches) increments downstream of the thermocouple, respectively.

Steady-state thermal efficiencies were derived from steady-state flue gas temperature and CO_2 concentration data according to a table of values given in Ref. 6. The tabulated relationships are plotted in Fig. 8 as a family of curves. During cyclical operation in which steady-state was not reached, values for those parameters just prior to burner cutoff were used in the same manner to get approximations of steady-state efficiencies. Burner firing times of 10 minutes gave such pseudo-steady-state efficiencies which were indistinguishable from those derived from steady-state measurements; those calculated from 4-minute burner firing time data were approximately 1/2 to 1% higher than the steady-state efficiencies.

Determination of furnace thermal performance during cyclical operation is more difficult than during steady-state operation. To avoid the complications of measuring or estimating transient draft air and

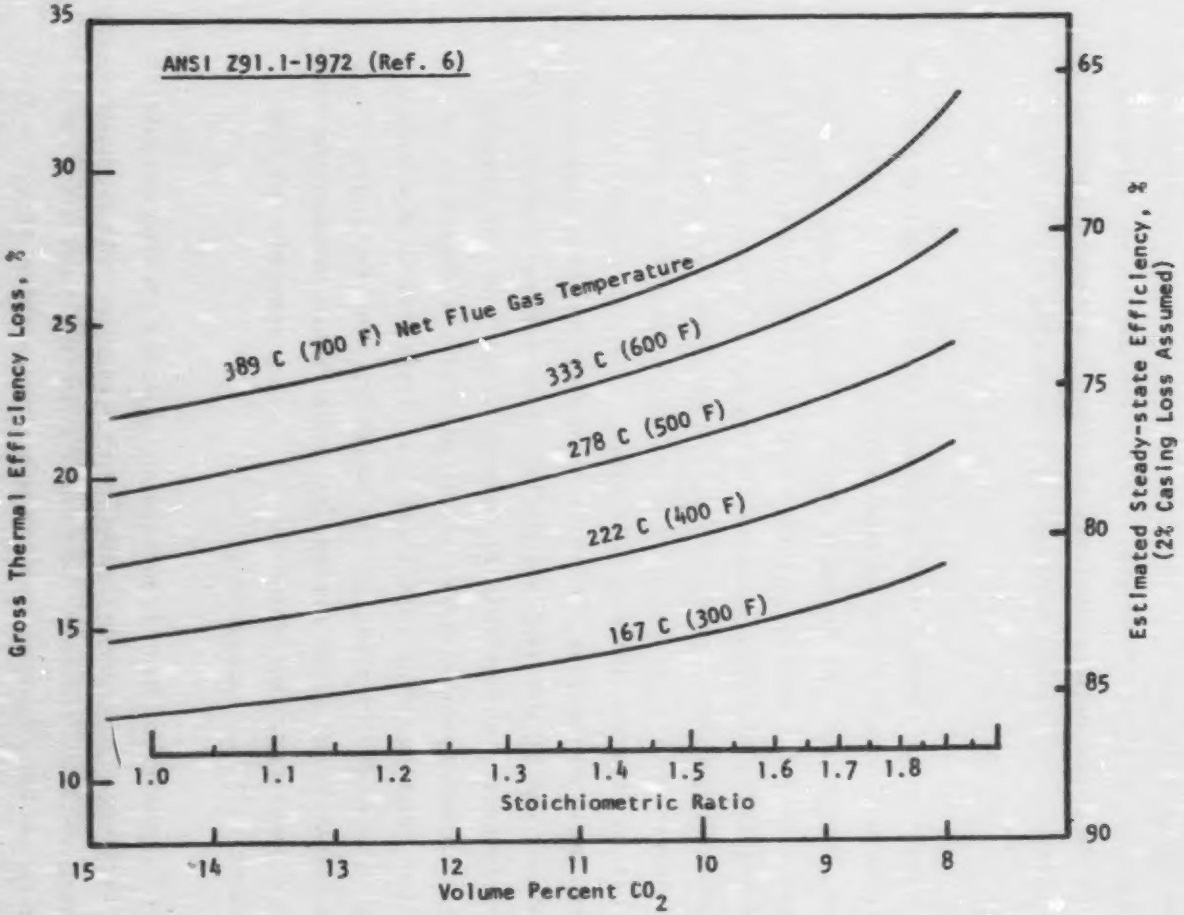


Figure 8. Flue Gas Gross Thermal Efficiency Losses as a Function of Net Flue Gas Temperature and Composition

furnace cabinet heat losses, the method used to determine cycle-averaged efficiency was to divide the warm-air furnace coolant net heat gain by the gross heat input of the fuel burned in a cycle. This required measurements of oil flowrate, oil and combustion air temperatures and, for the warm-air furnace coolant, flowrate and temperatures at the inlet and outlet. Provisions were made for measuring all of those parameters.

The inlet warm air was drawn into the furnace from the ambient outdoor atmosphere through a 0.46 m (18 inches) square duct with an inlet flair and internal "egg-crate" flow straightener. The volumetric air flow was measured with a cumulative readout, gas-flow anemometer, i.e., it integrated the total air flow admitted during each complete cycle. Ambient atmospheric pressure, temperature, and relative humidity were recorded continuously at a meteorological data station located approximately 15 meters from the furnace test stand. Furnace coolant air temperatures were measured at the inlet aneometer location with a mercury therometer and at the warm-air outlet as an average reading from nine, ice-referenced, chromel/alumel thermocouples in a rectangular grid array. In addition, an array of six more of the same type of thermocouples (shielded from radiation) was installed in the warm-air passageway between the firebox and the secondary heat exchanger. This provided a measurement of the average air temperature between the firebox and heat exchanger, from which the heat being removed from the finned firebox could be calculated.

A variety of combustion, heat transfer, and thermal cycling parameters were also recorded. Six chromel/alumel thermocouples were attached to the combustor and heat exchanger sections to monitor peak metal temperatures for an estimation of thermal stresses and service life. A dial thermometer was used to measure temperatures in the sealed vestibule area to determine the likelihood of falling outside of code specifications or of accelerated component degradation from anticipated higher temperatures in the sealed compartment. Manometer taps in the combustor section and in the heat exchanger outlet manifold were used to measure "over fire" draft and heat exchanger pressure drop. Calculated adiabatic flame conditions and the warm-air temperature rise through the finned combustor section were used to estimate the combustion gas temperature at the exit of the combustor.

Relative pressure measurements were taken at the warm-air blower outlet, at a point between the combustor and heat exchanger, and at the warm-air outlet. The outlet back pressure could be varied through a set of adjustable outlet louvers to simulate various installed ducting resistances. The warm-air flowrate through the prototype furnace also could be varied independently through use of a 3-speed fan control and a variable-spacing (effective diameter) drive pulley.

The electrical consumption of the components could be measured individually by a 1000-watt, alternating-current wattmeter, while the components were operating at their respective design conditions. The measured electrical power consumption then could be added to the fuel's gross energy input to obtain total energy consumption figures.

TEST RESULTS

The methodology used in the experimental testing was to proceed, more or less sequentially, through the test matrix presented in Table 1 with considerations for flexibility as problems or promising discoveries arose. Basically, the test matrix was intended: (1) to establish baseline emissions and thermal performance characteristics of the stock furnace prior to its conversion to a prototype unit, and (2) to optimize the operation of the prototype optimum furnace. Complete tabulations of the data obtained are contained in Appendices B and C. Data were recorded for 401 runs. Due to alternating laboratory effort with another related study (Ref. 2) and elimination of some checkout tests, there are some discontinuities in the sequence of run numbers.

Stock Lennox Furnace

The Lennox Model 011-140 furnace, described earlier, was installed in the laboratory test facility (Fig. 7), and tests were conducted to measure its air pollutant emissions and thermal efficiency characteristics.

Air Pollutant Emissions - The Lennox furnace was fired in its stock configuration with the factory-supplied 0.85-70°-A oil nozzle*. Cycle-averaged pollutant emission data are given in Appendix B, Table B-1. The stock furnace operated at a nominal stoichiometric ratio of ~1.60, which is fairly typical of new oil heating equipment presently being produced. The cycle-averaged Bacharach smoke readings were No. 1 or greater due to high readings obtained immediately after burner start. The recovery to zero smoke on most of the runs occurred quickly, within

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*Oil nozzle callouts designate the nominal fuel flowrate, the fuel spray cone angle, and the general type of spray configuration. For example, this "0.85-70°-A" callout denotes a firing rate of 0.85 gph (0.89 ml/s) and a hollow-cone-type spray ("A") with a 70° cone angle. Spray nozzles which produce solid-cone sprays carry a "B" designation.

TABLE 1. FURNACE SYSTEM EVALUATION TEST MATRIX

Furnace Unit	Firing Rate, ml/s	Warm-Air Flow, m^3/s	Stoichiometric Ratio Range	Cycle Time On/Off, minutes
Stock (Lennox)	0.89	Manufacturer's Recommended Value	1.40-1.60	Steady-State
				4/8
				8/4
				10/20
				Steady-State
				4/8
				8/4
	↓	↓	↓	10/20
Optimized Prototype	1.05	0.708	1.10-1.20	Steady-State
				4/8
		0.637		
		0.566	↓	↓
		Optimum	Optimum	Steady-State
				4/8
				8/4
	↓	↓	↓	10/20
	0.89	0.566	1.10-1.20	Steady-State
				4/8
				8/4
	↓	↓	↓	10/20
	Optimum	Optimum	Optimum	Long-Term Cyclic Operation

the first half minute, and these runs are flagged by asterisks in Table B-1. This was considered to be acceptable operation due to the quick recovery and also because the averaging method gave excessive weight to the start conditions. Taken once every minute, each smoke reading was assumed to be a representative average for a full minute. The initial reading, taken during the short high-smoke interval, was not an accurate representation of the first minute average, and the magnitude of the error was emphazied by the short (4 minutes) burner firing times.

Carbon monoxide and unburned hydrocarbon emissions were acceptably low at all operating conditions tested. Nitric oxide emissions, plotted in Fig. 9, generally exceeded 2.0 g NO/kg fuel burned*; they were higher than anticipated for the 0.89 ml/s firing rate, based on experience with other residential burners (Ref. 1). The flue gas temperatures were quite low, on the order of 180 C (~360 F), consistent with the 0.89 ml/s (0.85 gph) firing rate being the minimum recommended for this furnace.

The stock furnace was then fired with a 1.00-70°-A (1.05 ml/s) oil nozzle. It appeared to light-off better at this higher firing rate, seen as an improvement in the cycle-averaged smoke emissions, resulting in a lower operating stoichiometric ratio range (~1.40-1.50) than the 0.89 ml/s firing rate. The other pollutant concentrations remained essentially unchanged with the nitric oxide (NO) level remaining nominally about 2.5 g NO/kg of fuel burned.

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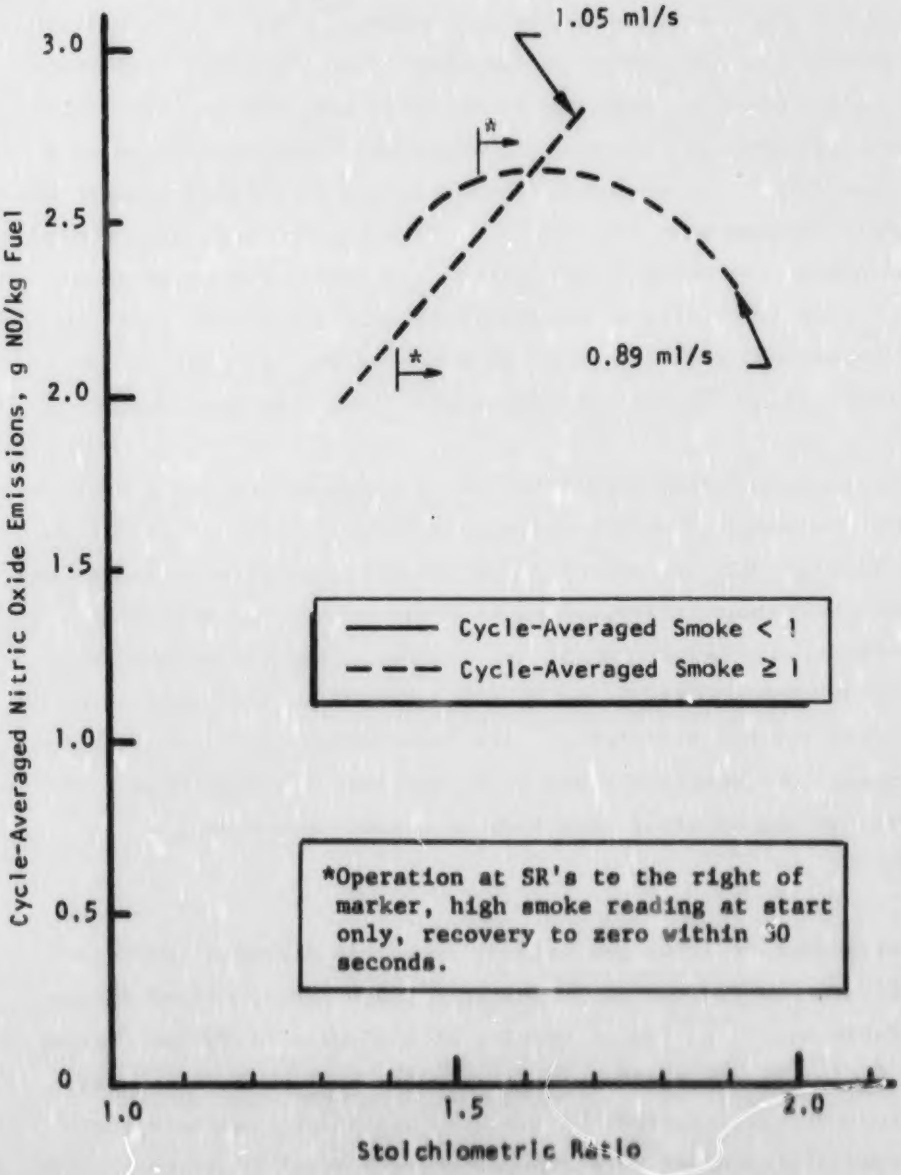
*Oxides of nitrogen emissions (NO_x) from residential heating systems have been observed (Ref. 4 and 7) as comprising NO (nitric oxide) and NO_2 (nitrogen dioxide) in volumetric proportions of about 9:1 to 10:1. Thus, on the average, NO accounts for more than 90 mole percent of NO_x emissions. For that reason, NO measurements were used throughout this investigation as a quantitative indicator of NO_x emissions, and NO_2 emissions were not measured.



*Operation at SR's to the right of marker, high smoke reading at start only, recovery to zero within 30 seconds.

Figure 9. Cycle-Averaged Nitric Oxide Emissions from the Lennox O11-140 Furnace in its Stock Configuration

Thermal Efficiency - Gross thermal efficiencies (based on the fuel's higher heating value) were derived by two methods. The first, referred to as the warm-air method, involved calculation of the total heat transferred from the combustion products to the warm air, and was based upon measured data for the warm air mass flowrate and temperature rise, and the total fuel input. In cyclical tests, because furnace components are often partially heated when the air is not flowing and partially cooled when the burner is not being fired, these parameters were measured as functions of time and averaged appropriately over the firing cycle to obtain cycle-averaged efficiencies. This method included all operational thermal losses (casing, standby draft flow, flue gas, etc.).

The second method, referred to as the flue gas method, was adapted from the method recommended in Ref. 6 for determining oil furnace efficiencies. The flue gas CO_2 concentration and temperature were measured, and an efficiency decrement corresponding to their values was read from Fig. 8. Subtracting that decrement and a modest casing loss (2% or less) for conduction and radiation to the surrounding from 100% gave the estimated gross thermal efficiency. The ANSI method is for application to steady-state; our adaptation was to assume that the last minute of burner firing during cyclical operation is a good approximation of steady state.

Estimated steady-state (flue gas method) and cycle-averaged (warm-air method) efficiencies for the stock Lennox furnace are tabulated in Appendix B, Table B-2. Results of testing at 1.05 ml/s (1.00 gph) firing rate (Runs 56 to 52) are plotted in Fig. 10 for both methods. Linear least-squares correlating lines fit to the data points indicate that cycle-averaged efficiencies were approximately 5 to 6% lower than those for steady state at comparable stoichiometric conditions. Nearly comparable 5% differences between steady-state and cyclical efficiencies were measured for two other warm-air furnaces tested earlier in this

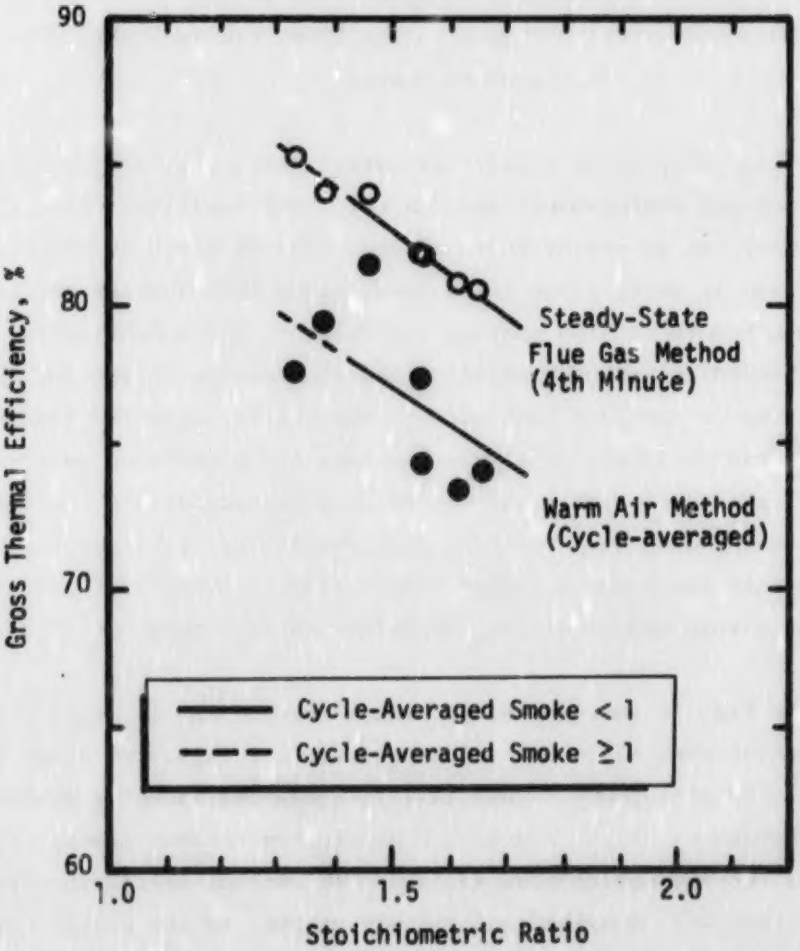


Figure 10. Thermal Efficiency Characteristics of a Stock Lennox O11-140 Warm-Air Oil Furnace Fired at 1.05 ml/s in 4-Min-On/8-Min-Off Cycles

facility (Ref. 2). Those units were fired with longer 30-minute (10 minutes on/20 minutes off) cycles than the 12-minute cycles (4 minutes on/8 minutes off) fired in the stock Lennox.

The data in Fig. 10 exhibit a characteristic observed in all cyclical furnace efficiency measurements made in this test facility, viz., the scatter is very low in steady-state efficiencies obtained by the flue gas method, and is quite large in cycle-averaged efficiencies obtained by the warm-air method. The testing was delayed, and a substantial effort was expended toward understanding and eliminating causes for the scatter. It was determined that much of the scatter resulted from testing outdoors and could not be eliminated (see the Discussion section). As a result, greater reliance was placed upon the steady-state efficiency values for comparing one burner or one furnace with another while recognizing that there was a larger undertainty in cycle-averaged efficiencies for situations requiring comparison through them.

It is seen in Fig. 10 that the stock Lennox furnace can be tuned for normal operation with 45 to 50% excess air where it achieves about 82 to 83% steady-state efficiency. This is well above the industry minimum performance standard of 75% and very close to the maximum practical limit of 85% (noncondensing flue gases). The compact heat exchanger appears to be very well designed and closely matched to the unit's burner and firebox. Net flue gas temperatures are as low as should be designed for, so the only way that steady-state efficiency could be improved is by lowering the burner's excess air requirement. A potential gain of only about 2 percentage points might be realized.

Optimum Burner in the Stock Furnace - The 1 ml/s (gph) optimum burner unit was installed and fired in the stock Lennox furnace. Test results (Runs 74-87, Tables B-1 and B-2) indicated that this burner, as compared with the stock Lennox burner, might offer modest improvements in both NO emissions and thermal efficiency: Measured NO emissions were reduced by about 5 to 7%, and smoke-free operation was possible with less than 27% excess air.

Prototype Optimum Furnace

Initial Shakedown Tests - A series of tests was made with the prototype optimum furnace system, as designed, operating at its nominal 1.05 ml/s (1.00 gph) firing rate conditions (Runs 149-154, Table B-3, Appendix B). Operationally, the system behaved reasonably well for its first test series. There were, however, some surprising results from the data analysis. The burner was not able to operate smoke-free at excess air levels below about 20%. Flue gas NO concentrations were 60 to 80% higher than the target value of 0.50 g/kg, even though the temperature measurements indicated that nearly 33% of the fuel's higher heating value was extracted from the finned combustion chamber (versus the design target of 20%). Additionally, thermal efficiency was lower than expected.

To investigate the cause of the higher excess air operational requirements and the higher NO emissions, the optimum burner was temporarily removed and replaced, successively, by two different flame-retention burners. The first was a Beckett Model AF burner, which was the stock burner for a Williamson Model 1167-15 furnace tested extensively in the Ref. 2 studies (and modified later in the current investigation). It was fired in the prototype furnace in Runs 155 to 162 (Tables B-3 and B-4). The second, the stock Lennox burner supplied with the stock Lennox Model 011-140 furnace, was tested in Runs 163-170 (Tables B-3 and B-4). Results obtained with these burners are described in the following subsections together with results from using the optimum burner.

Pollutant Emissions - Cycle-averaged emissions data concerning flue gas NO and the transition through No. 1 smoke are plotted in Fig. 11 for

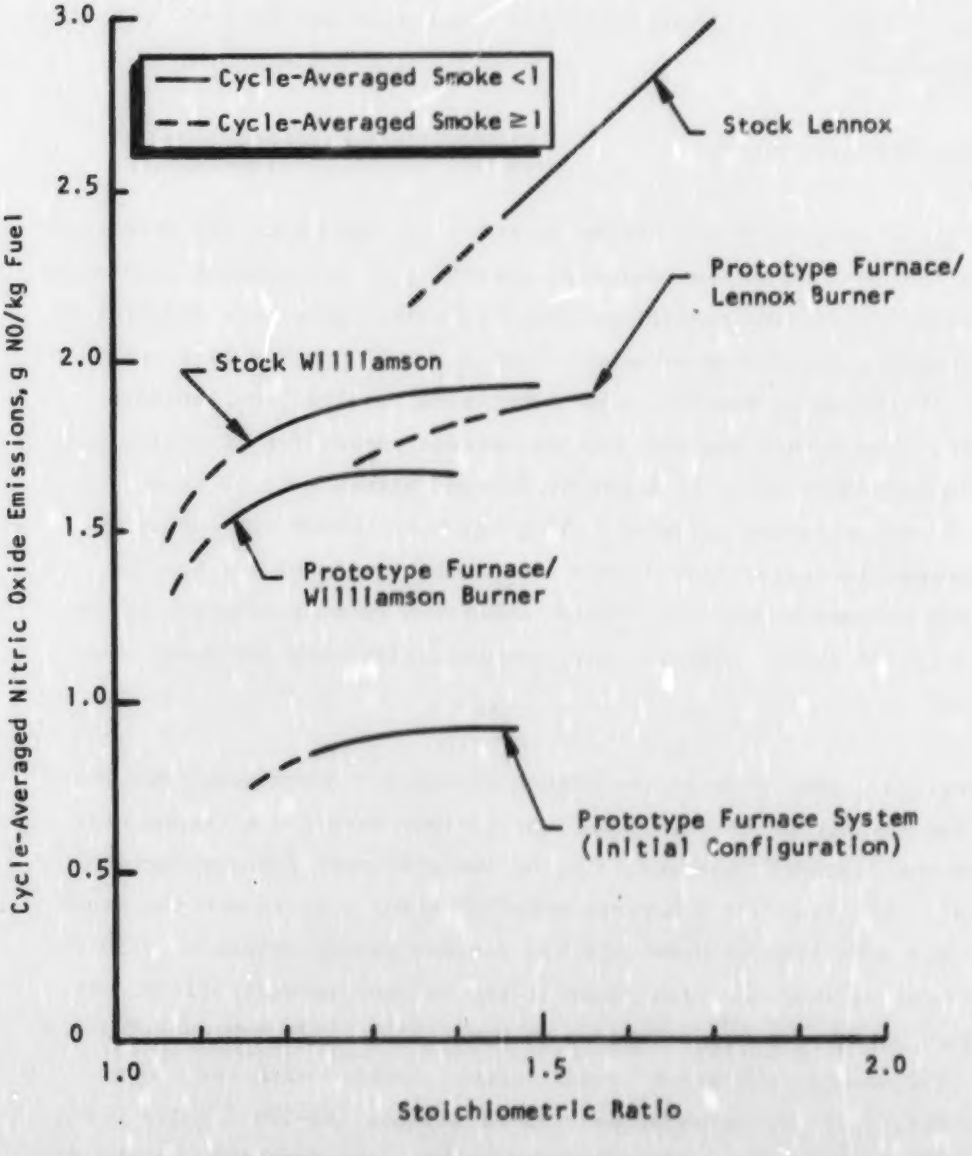


Figure 11. Comparison of Cycle-Averaged Nitric Oxide Emissions of Various Oil Burners in the Prototype Furnace and in Their Respective Furnaces

the prototype optimum furnace tested with all three burners mentioned above. Additionally, comparable data are shown for the two stock furnaces which were equipped originally with the flame-retention burners. Comparison of the curves in Fig. 11 shows that: (1) NO emissions produced by the optimum furnace with the flame-retention burners were about 15 and 25% lower, respectively, than those from the stock Williamson and Lennox furnaces; (2) flame-retention burners produced approximately twice as much NO when fired in the prototype furnace as did the optimum burner; and (3) firing the flame-retention burners in the finned air-cooled combustor, rather than in their stock furnaces' refractory-lined combustors, did not change appreciably the excess air level at which transition to No. 1 smoke occurred. Furthermore, there apparently was no degradation of other operational characteristics (e.g., light-off, transition to steady state, combustion noise) of the flame-retention burners. The optimum burner, on the other hand, exhibited appreciable low-frequency (rumbling) noise at the lower excess air levels tested.

It was thought that the operational behavior of the optimum burner might have been degraded by extraction of more heat from the flame zone than had been intended. Therefore, in an attempt to decrease combustion zone heat extraction and increase combustor wall temperatures, the warm-air flow was reduced from 0.57 m^3/s (1200 cfm) to 0.46 m^3/s (950 cfm) for a series of tests (Runs 171-176, Table B-3). Rather than helping to broaden the operational envelope for the prototype furnace, this change seemed to emphasize the tendency toward noisy combustion. At the 22% excess air level, no smoke was formed but operation was marginal due to intermittent roughness; smooth operation was not attainable at any lower excess air levels.

Thermal Efficiency - Steady-state thermal efficiencies for the prototype furnace fired with the optimum burner and with the two flame-retention burners from the stock Lennox and Williamson furnaces are tabulated in Table B-4 and are plotted in Fig. 12. The steady-state efficiency curve for the stock Lennox furnace (Fig. 10) also is reproduced

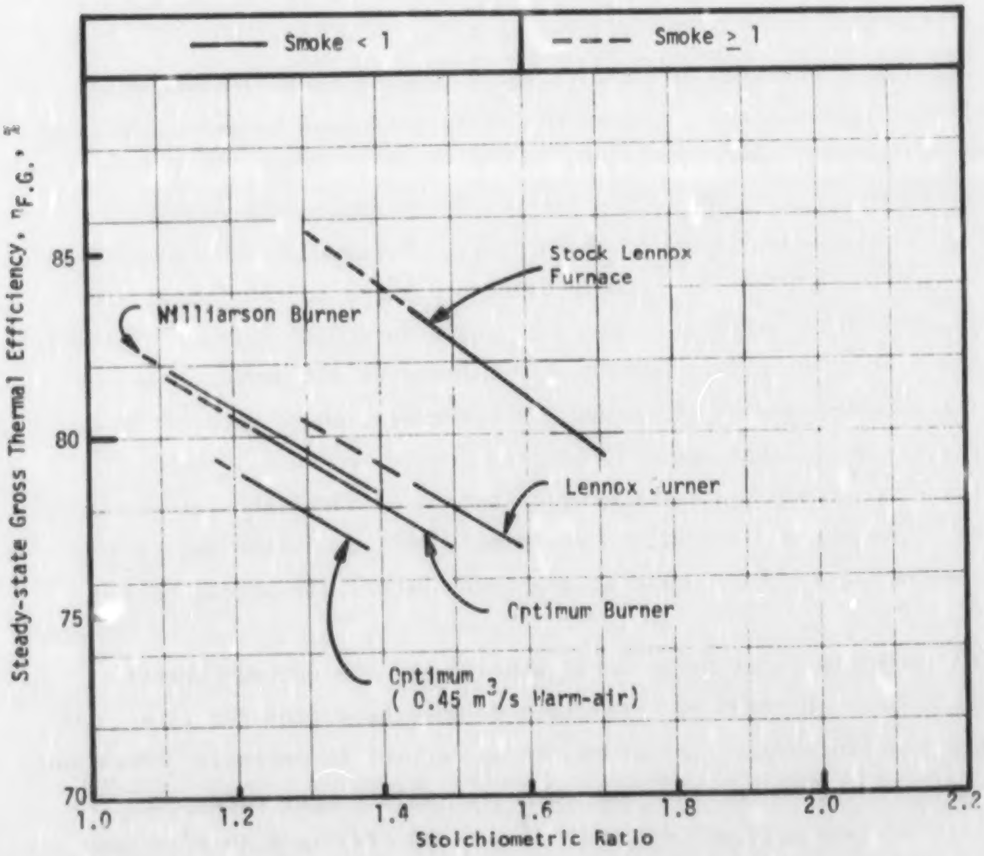


Figure 12. Comparison of Thermal Efficiency Characteristics of the Prototype Optimum Furnace With Various Burners and System Changes at a Nominal Warm-air Flowrate of 0.566 m^3/s

on this graph. It was noted above that about 33% of the fuel's higher heating value was transferred to the warm-air furnace coolant through the finned firebox walls. In spite of this substantial supplemental heat transfer, the optimum furnace's steady-state efficiency was 4 to 5 percentage points lower than that of the stock Lennox furnace when they were fired with the same Lennox burner and at equivalent stoichiometric ratios. Its efficiency with the optimum burner at comparable operating conditions was even lower by an additional 1 percent, and was degraded further by 1 percent or more when combustor cooling was reduced at the lower warm-air furnace coolant flowrate.

The observed drop in thermal efficiency when the stock furnace was converted to the prototype furnace was suspected initially to be caused either by incomplete combustion in the firebox, resulting in some delayed heat release in the heat exchanger, or by the optimum burner somehow producing very nonuniform, thermally striated flow through the heat exchanger. If these phenomena existed, they should have been substantially eliminated by the flame-retention burners, since such burners usually mix the incoming reactants more vigorously and burn them more quickly than do conventional burners. In other words, one or both retention head burners should have restored most of any efficiency drop caused by slow combustion. The fact that they did not do so was interpreted as meaning that the efficiency problem was probably caused by design changes in the warm-air side, rather than the combustion gas side, of the furnace's heat transfer process.

Testing With Modified Components - It was evident from the results of the shakedown tests described above that the emissions and performance goals could not be met by the prototype furnace without some refinement of its initial design. Achieving the emissions goals obviously would require some improvements in the combustion equipment design. Achieving the efficiency goal, on the other hand, required improvements in the heat transfer equipment, predominantly in the warm-air circuit. As discussed later in the description of the final design, it was believed that heat absorption by the warm air could be corrected by redesigning the baffles in its passages and, if necessary, could be supplemented easily by increasing the secondary heat transfer area. As a result, the subsequent testing was devoted almost exclusively to what was considered to be the more difficult task of improving the combustion circuit to achieve the emissions goals.

The approach taken was to investigate the effects on acceptable operating conditions and pollutant emissions of specific design differences between the prototype furnace and the research combustor apparatus (Ref. 3) from which the design criteria were derived. Because the optimum burner used in the two systems was the same, its design was not varied during a large portion of the subsequent testing. Instead, attention was focused on the finned air-cooled firebox, the transition between it and the main heat exchanger, and the heat exchanger design and cooling media. The results of testing configurational variations in these components were very informative, but it appeared that applying them to achieve the emissions goals would make the furnace appreciably more complicated and expensive to manufacture. Attention was redirected, therefore, to determine whether some further refinement of the burner design might be beneficial. Experiments in that direction were performed with the prototype furnace in its initial design configuration; for that reason, the results are tabulated in Appendix B along with those from the initial shakedown tests. Those results are described next, before describing results from the intervening furnace modification tests.

Refined Burner Design Tests - The optimum low-emission burner was modified, as described on page 19, to reduce its potential for coupling with and amplifying oscillatory or noisy combustion. A quiet stator plate was installed on the discharge side of the combustion air fan, and a large-diameter static disc was installed inside the blast tube. Additionally, some tests were run to determine the optimum oil spray angle. The modified burner was tested in Runs 469 to 481 and 512 to 533 (some of which also had an internal baffle added to the firebox or heat exchanger). Emissions and performance data are tabulated in Appendix B, Tables B-7 and B-8, respectively.

With the 90-degree oil spray angle, the burner design changes actually were detrimental; at comparable excess air conditions, flue gas temperature and smoke and NO_x emissions were all increased (Runs 462-465 versus

Runs 452-457). With the smaller 60- and 70-degree spray angles, however, these conditions were all improved, and the furnace could be operated consistently at design goal conditions of only 15% excess air. Emissions of NO were slightly lower with the 1.0-70°-A nozzle, so it was designated as the preferred nozzle for the refined design burner.

The NO emissions achieved by this refined optimum burner in the prototype furnace came close to but remained above the 0.5 g NO/kg goal. A linear least-squares correlation of the data from Runs 469-477 ran from 0.51 g NO/kg at a smoky stoichiometric ratio (S.R. = 1.00) to 0.78 g/kg at S.R. = 1.35; tuned to S.R. = 1.15 as a nominal design point, 0.63 g NO/kg fuel burned would be produced.

Thermal efficiencies in that series of tests were slightly higher than those measured in the initial shakedown tests. Referring to Fig. 12, they were about midway between the lines for the Lennox and Williamson burners, i.e., about 1% higher than with the optimum burner.

A series of four tests (Runs 478-481) was made in which the warm-air blower was cut off immediately after burner cutoff to see if this would further reduce the carbonaceous emissions. Although the CO emissions were almost unchanged, both the UHC and smoke levels were increased. Therefore, this change in furnace control method is not recommended.

In view of the improved operational capability and lower NO emissions achieved by refining the burner design, an effort was made to further reduce combustion noise by adding sound-absorbing material inside the firebox and heat exchanger. A 0.01 m (3/8 inch) thick layer of pyroflex refractory fiber material was bonded to the dished bottom of the finned firebox and another was bonded to the flat top of the central dome portion of the heat exchanger. Test results (Runs 482-485, Table B-7) showed that NO production was increased and that smoke levels were unacceptably high over the entire stoichiometric range of interest.

Obviously, the layer of insulation on the bottom of the firebox had altered the flame zone recirculation and heat extraction characteristics drastically, so it was removed for the next series of tests (Runs 486-490). This restored the NO emissions to the former level. However, at comparable excess air levels this configuration produced more smoke and UHC than without any pyroflex lining.

Long-Duration Test - A simulated service test was conducted by running the prototype optimum furnace for a period exceeding 500 hours of cyclical operation. The furnace was adjusted initially to operate at approximately 15% excess air (design point), and set to fire automatically on a 4-minute-on/8-minute-off duty cycle. Reduction of initial performance data showed the actual excess air to be 17% (Run 529, Table B-7). The furnace was left unattended for the entire duration of the test with only general, external visual inspections made periodically. At the end of the 500-hour test, inspection of the furnace and burner revealed no signs of deterioration in any components. The excess air level had not changed appreciably from the initial 17% excess air setting.

Cycle-averaged flue gas concentrations of pollutant species were measured at the beginning and termination of the 500-hour test. There were slight increases in the emission levels of carbonaceous pollutants: smoke from 0 to an estimated 0.1*, UHC from 0.035 to 0.041 g/kg fuel, and CO from 0.35 to 0.54 g/kg fuel. Together with a slight increase in flue gas CO_2 concentration, these data indicate that the excess air level probably decreased slightly, in contrast to the indicated constancy from the measured CO_2, O_2, etc., data. The measured flue gas concentration of NO corresponded to 0.64 g NO/kg fuel burned, agreeing precisely with the least-squares correlation cited earlier. There was less than 1% difference between the values measured at the beginning and at the end of the 500-hour test.

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*The Bacharach smoke readings after 500 hours of service showed a very faint, almost imperceptible shade of gray that was constant through the firing period (not a start spike). As faint as this reading was, the filter was not snow white as it was in the initial readings; therefore, a nominal cycle-averaged value of 0.1 was assigned to the smoke reading.

Exploratory Modifications of the Prototype Optimum Furnace Configuration

Following the shakedown tests of the as-designed prototype optimum furnace, in which it was found not to operate satisfactorily with less than approximately 20% excess air and to produce substantially more than the target NO emissions, a number of exploratory modifications of the furnace configuration were investigated.

A close visual examination of the internal geometry of the firebox and heat exchanger revealed that the large rectangular entrance to the rear manifold was very close to the firebox. It appeared that the combustion gas flow might enter the heat exchanger too quickly, thereby producing smoke from premature quenching of combustion reactions. Therefore, a large baffle was installed above the firebox that blocked half of the central vertical cylinder, attached at the bottom of the rectangular entrance to the heat exchanger and canted forward at a 45-degree angle. The results (Runs 198-202, Table B-5) showed no significant improvement in the pollutant emissions characteristics.

When that proved not to be beneficial, the optimum furnace's combustion system was compared critically with the research combustor apparatus which had been used in the experimental derivation of the design criteria forming the basis for the optimum furnace design (Ref. 3). Design differences were identified which conceivably may have caused the observed differences in their achievable operating conditions and emissions. Furnace modifications made to assess the validity of hypothesized causes and of potential solutions were in the general areas of the firebox and the main heat exchanger.

Firebox Design Differences - One hypothesis for explaining the higher than expected NO results (0.85-0.90 g/kg) was that the eccentric reduction in diameter (where the combustion chamber is joined to the air-cooled heat exchanger) may have promoted greater recirculation in the combustor and produced a significant increase in mean gas residence time. To test that hypothesis, an available Williamson Model 1167-15 furnace was modified (see Fig. 13) by replacing its combustor with the smaller-diameter (0.254 m inside diameter), straight-cylindrical, pre-prototype, finned, air-cooled combustor tested earlier as a research combustion chamber (Ref. 3, Appendix F). That chamber was shortened to fit within the Williamson furnace cabinet, and was welded to the Williamson air-cooled heat exchanger. This modified furnace is referred to as the "finned-combustor Williamson" furnace. It was tested with the optimum 1 ml/s (gph) burner.

Flue gas NO emissions measured with the finned combustor Williamson furnace (Appendix C, Table C-1) are presented graphically in Fig. 14 along with earlier experimental results from tests of the finned-combustor in the prototype optimum furnace and from tests of the two stock furnaces from which these finned-combustor furnaces were derived. The NO emission characteristics of the finned-combustor Williamson furnace are seen to be more like those of the finned-combustor optimum furnace than of the previous research setup tests of its finned combustor. In fact, comparison of the several curves in Fig. 14 strongly suggests that the larger-diameter (0.305 m) finned-combustor in the optimum furnace was appreciably better with respect to NO and smoke emissions than the one in the modified Williamson furnace. Therefore, the hypothesis that uncooled combustion gas recirculation (i.e., longer residence times) in the larger firebox caused the higher-than-anticipated NO emissions from the prototype furnace did not appear to be valid.

Heat Exchanger Design Differences - In each of these experimental furnaces, there was a closed-top cylindrical dome above the combustion

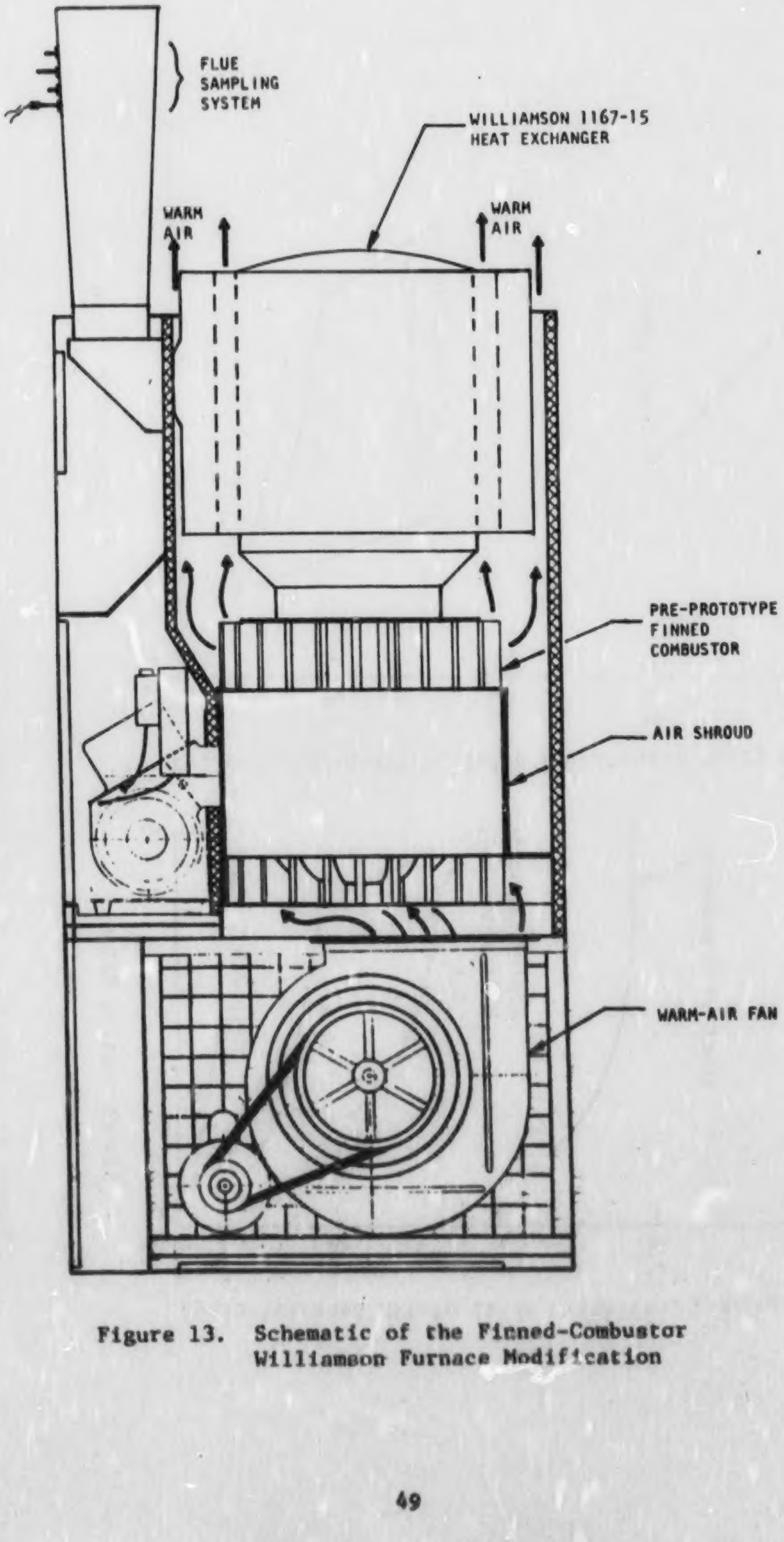

FLUE SAMPLING SYSTEM
WILLIAMSON 1167-15 HEAT EXCHANGER
WARM AIR
WARM AIR
PRE-PROTOTYPE FINNED COMBUSTOR
AIR SHROUD
WARM-AIR FAN

Figure 13. Schematic of the Finned-Combustor Williamson Furnace Modification

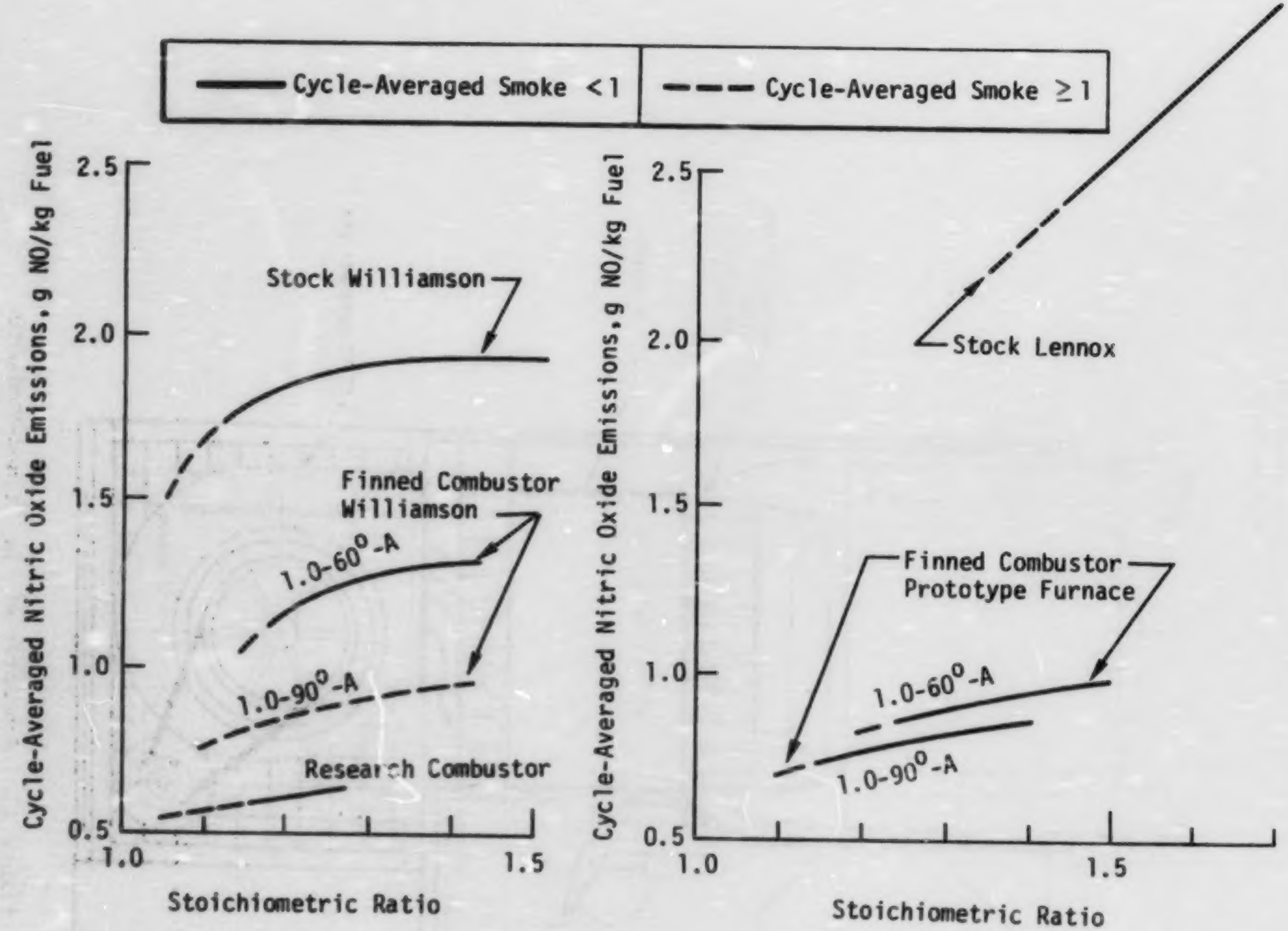


Figure 14. Comparison of Cycle-Averaged Nitric Oxide Emission Characteristics of Two Finned Combustor/Air-Cooled Heat Exchanger Furnaces

chamber. The combustion gases flow up into the dome and out through a rectangular opening in one side wall into the furnace's air-cooled heat exchanger. In contrast, the research tests of the finned combustor utilized a long, vertically disposed 0.25 m (10 inches) diameter pipe above the finned combustor, with a water-cooled copper coil heat exchanger suspended in the combustion product gases. Thus, it appeared that there were two significant design differences which might be responsible for the unexpectedly high NO and smoke conditions, viz, water cooling versus air cooling and side discharge versus vertical updraft discharge. These differences were tested sequentially.

To test water-cooling versus air-cooling effects, the prototype optimum furnace was modified. The air-cooled heat exchanger was removed from the furnace assembly and a 0.25 m (10 inches) diameter vertical pipe with the water-cooled spiral coil was installed in its place (see Fig. 15). This is referred to as the "coil-cooled" prototype furnace. Steady-state NO emissions measured during several series of experiments (Runs 221-303, Table C-2) with this experimental furnace configuration are plotted in Fig. 16; variations were made in the coil position and its exposure to the flame zone and the coolant passed through the coil. Notations assigned to the curves of Fig. 16 refer to: (1) heat exchanger position, $0.30 \leq L \leq 0.75$ m from the inside bottom of the combustor to the bottom of the heat exchanger coil; (2) flue gas temperature downstream of (above) the coil; and (3) the presence of a stainless-steel radiation shield 0.05 m below the heat exchanger coil.

The design of the finned-combustor for the protytype furnace was based heavily upon data obtained earlier in research combustors with the water-cooled coil heat exchanger positioned at $L \geq 0.50$ m (Ref. 3). The bold-face L = 0.50 m, water-cooled curve in Fig. 16 is of great interest because it duplicates exactly the NO and operational characteristics that had been anticipated for the prototype combustor design. The NO emissions are ≤ 0.50 g/kg at very low excess air ($\leq$12%), and zero smoke is produced at these conditions.

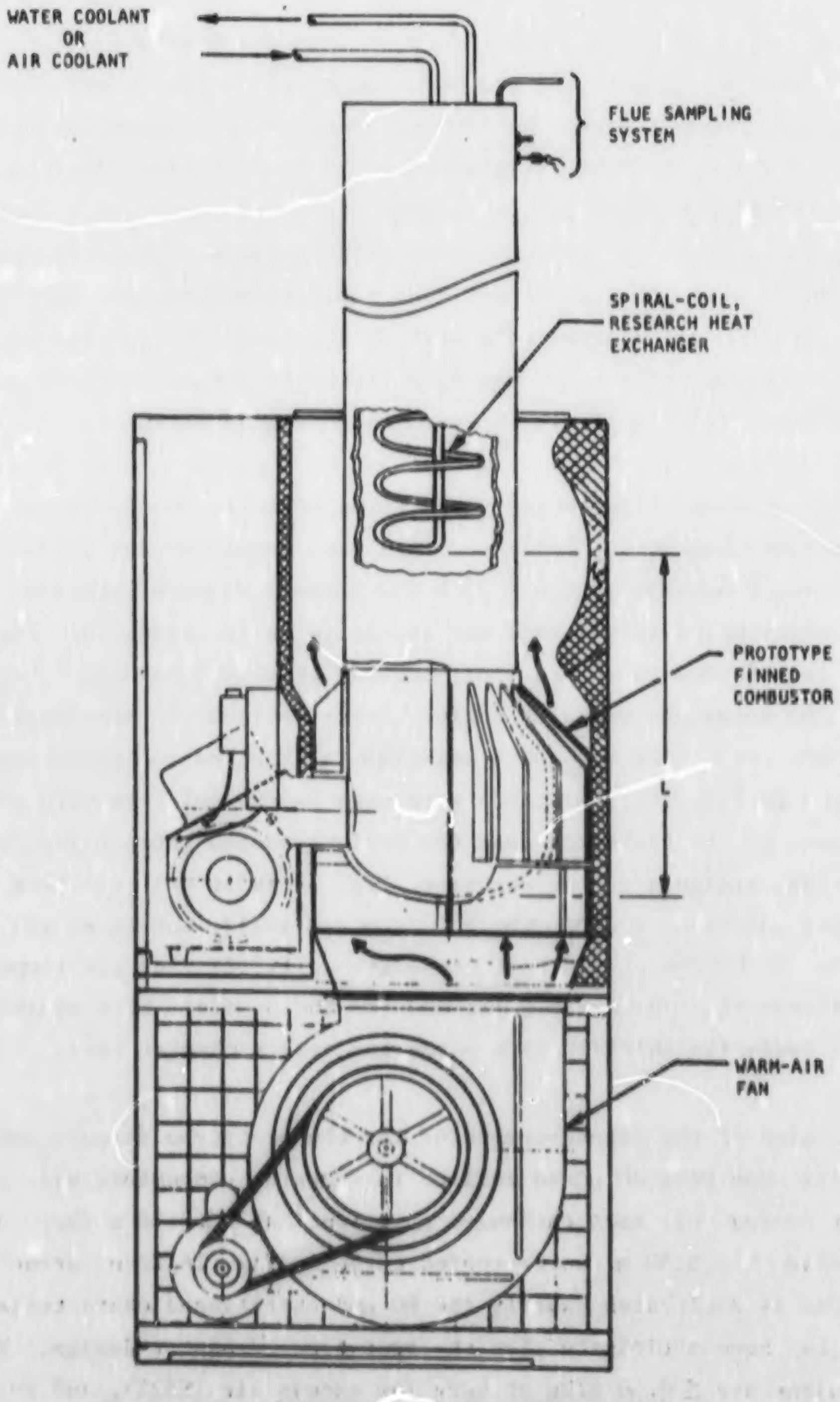

WATER COOLANT
OR
AIR COOLANT
FLUE SAMPLING SYSTEM
SPIRAL-COIL, RESEARCH HEAT EXCHANGER
PROTOTYPE FINNED COMBUSTOR
L
WARM-AIR FAN

Figure 15. Schematic of the Coil-Cooled Modified Prototype Furnace

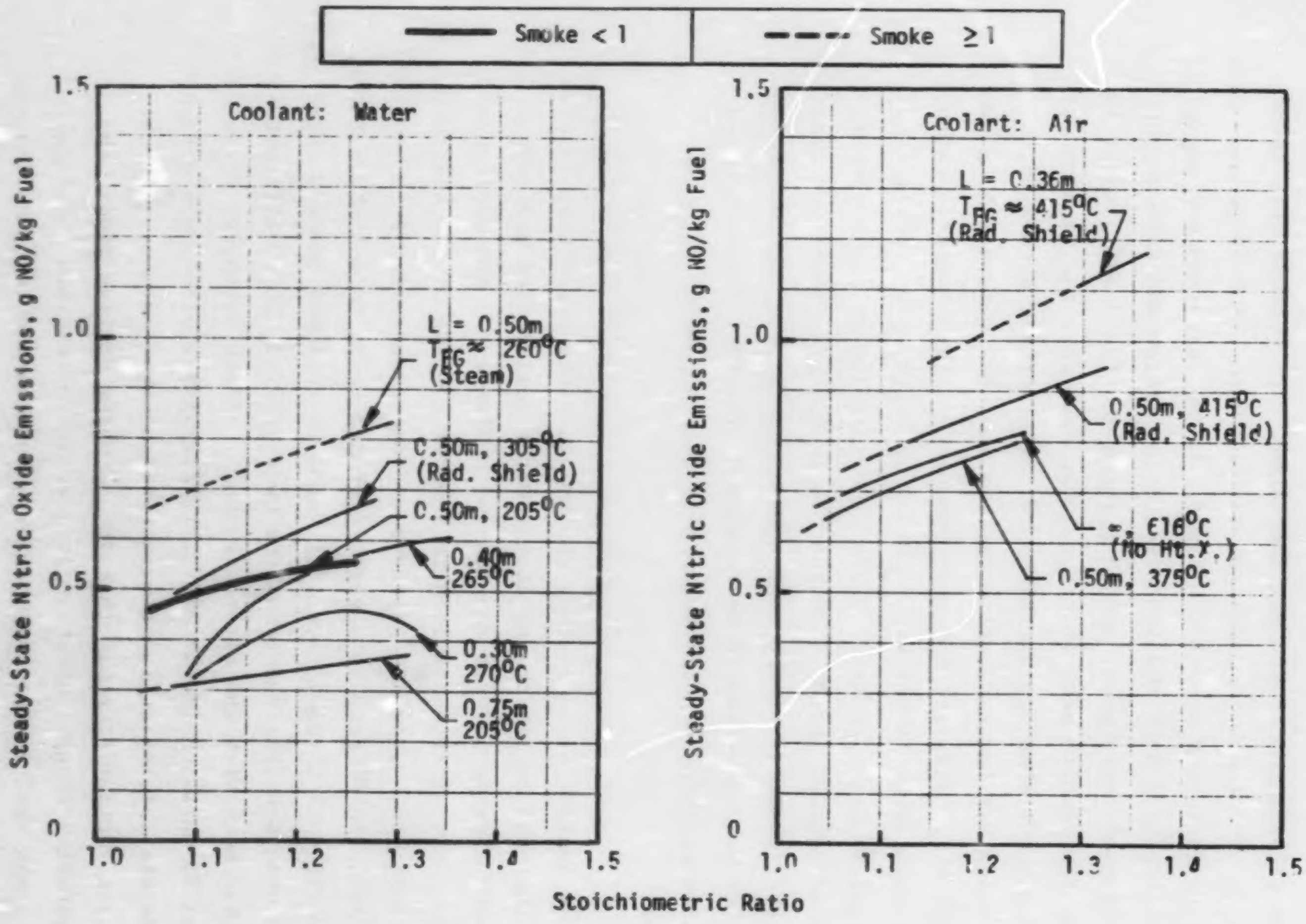


Figure 16. Steady-State Nitric Oxide Emissions From the Coil-Cooled Prototype Furnace With Various Coolant Media and Configuration Changes

As the water-coil heat exchanger was moved from L = 0.50 m to L = 0.30 m, an orderly family of curves was indicated, with successively lower NO. However, when the heat exchanger was moved to L = 0.75 m, some very unusual characteristics appeared. First, the NO curve dropped unexpectedly to ~0.35 g/kg. Both the instrumentation system and furnace operation were checked thoroughly, and no problem was found which would indicate that the results were in error. The next day, the furnace was started up without changing anything and, after it was warmed up, a series of tests was begun to again recheck the NO versus S.R. curve for L = 0.75 m. Initially, for three values of stoichiometric ratio, the NO emissions were in the 0.6 to 0.7 g/kg range, i.e., where they'd been expected to be, *a priori*. While running a fourth stoichiometric ratio, the NO dropped abruptly to the 0.3 to 0.4 g/kg level and remained there for the remainder of the test series. This behavior was not understood, but was suspected to be related to internal combustion gas recirculation patterns of different sorts that happened to become established.

To gain some insight as to whether the NO and smoke emissions, so dramatically improved by the change to the water-coil-cooled prototype furnace, were altered more by the cooling medium or by the configuration change, several retreats from full water cooling of the coil were tested. First, the water was replaced by compressed air. At L = 0.50 m (see Fig. 16), the NO emissions were increased by 30 to 40% above those from the water-cooled case. Next, the cooling coil was removed entirely, and it was observed that the emissions were only slightly different from what they had been with the air-cooled coil. (Interestingly, these 0.7 to 0.8 g/kg NO emission levels are quite comparable with those obtained in the same stoichiometric range with the prototype furnace, Fig. 11. This suggested rather strongly that it was water cooling alone, and not the configuration change, which influenced the NO emissions.) Finally, water was again used as the coil coolant, but its flowrate was reduced until the coil effluent was entirely gasified (steam). As expected, the NO emission results (labeled "steam" in Fig. 16) were identical with those measured with the air-cooled coil. Surprisingly, however, the flue gas became smoky, producing significantly greater than No. 1 smoke at conditions with less than 25% excess air. Neither the CO nor UHC concentrations were unusually high during these smoky conditions. Again, all systems were checked for indications of malfunctions and none were found. In fact, it was observed that simply turning up the water flowrate to the coolant coil eliminated the smoky flue gas condition.

To place the foregoing data in perspective, it is appropriate to recall that approximately 40% of the heat extraction from the flue gases was effected by forced convective air cooling of the finned combustion chambers. Thus, the mean temperature of the gases leaving one of these combustors was considerably lower than the adiabatic flame temperature which may be experienced at some portions of the flame zone. For example, at 20% excess air and 85% overall thermal efficiency; 40% heat removal would reduce the mean temperature from about 1830 C (3330 F) to 1080 C (1970 F). Such large temperature reductions should very effectively quench the kinetically limited production of NO within the combustion chamber. Therefore, it was considered that one or more of the following hypothese must accout for (contribute to) the demonstrated influence of the further-downstream, water-cooled, coil heat exchanger on flue gas NO:

1. There is highly striated flow out of the combustor, with very well-cooled gases near the chamber walls, and high-temperature, NO-producing gases forming a central core flow. Penetration of high-temperature striations into the heat exchanger tends to distribute the quenching of the gases along the flowpath length resulting in slower quenching, overall.
2. The flame zone, "seeing" the water-cooled coil, radiates sufficient additional heat to it to reduce peak flame zone temperatures appreciably, thereby lowering NO production rates.

3. The copper tubes of the coiled heat exchanger act catalytically to influence flue gas NO concentrations.

4. The heat exchanger influences NO production by inhibiting or promoting vertical combustion gas recirculation patterns, with stronger recirculation of cooler gases induced by the water-cooled coil than by the various air-cooled heat exchangers.

Indications of highly striated flow were sought by measuring the radial variations of flue gas temperature and NO concentration at the top of the 0.25 m diameter combustion chamber extension pipe when the coil heat exchanger was removed. Rather minimal radial variations (~10%) were observed, so striated flow was thought to be an unlikely contributor to the heat exchangers' influence on flue gas NO concentrations.

Radiant heat transfer rates from a high-temperature gas to a cool surface are proportional to the effectively seen surface area and to the difference between the fourth powers of their absolute temperatures, i.e., $\Delta(T^4) = T^4_{gas} - T^4_{surf}$. The flame radiates to the combustor wall as well as to the heat exchanger. It is instructive to consider the relative contributions of these two components to flame zone cooling. If $T_{gas} = 2000$ K, $T_{combustor} = 550$ K, and $T_{coil} = 300$ K, the $\Delta(T^4)$ driving potential for radiating heat to the "cold" coil is only about 1/2% higher than that for the combustor walls; i.e., to the flame, the chamber is also quite "cold." Further, the inside surface area of the combustor is at least four to five times the coil area that the flame can possibly view. Thus, there is little likelihood that radiation to a water-cooled coil cools the flame appreciably more than does radiation to a somewhat warmer air-cooled coil. Nonetheless, some experiments were conducted with a flat 0.18 m diameter stainless-steel "radiation shield" installed 0.05 m upstream of the leading coil of the 0.15 m diameter coil heat exchanger. In these tests, the radiation shield, rather than the first coil, was spaced the distance, L, above the bottom of the firebox. Three of the curves in Fig. 16 represent data obtained with that shield in place. In the tests at L = 0.50 m, flue gas NO was about 10% higher with the radiation shield than without it for both water and air as coil coolants. At least a portion of that increase is believed to have resulted from the slight downstream displacement of the coil. The NO level with the radiation-shielded, water-cooled coil remained well below the air- and steam-cooled levels, so these data were interpreted as refuting the hypothesis that the water-cooled coil affects NO production by radiant cooling of the primary flame zone.

While the radiation shield was in place, the air-cooled coil was lowered so that the radiation shield was located within what would be the short 0.25 m diameter cylindrical connection between the finned combustor and commercial air-cooled heat exchanger. What was sought was an indication of how the prototype furnace might perform if such a constricting plate were installed to restrict vertical recirculation of gases from the central closed-top dome back down into the combustion chamber. There resulted a sharp increase in NO emissions to above 1 g NO/kg and excessive smoke at 30% excess air and lower. This concept was an obviously detrimental one.

If copper were a catalyst for some NO reduction reaction, its effectiveness should be enhanced by moderate surface-temperature increases. Thus, the observed decrease in NO production with decreasing coil temperature seemed like the wrong direction for copper to be catalyzing an NO consumption reaction. As a final check, research combustor test data reported in Ref. 3 were reviewed, and cases were found where NO emissions obtained with the water-cooled copper coil were comparable with those produced by a water-cooled steel coil (which the copper coil had replaced).

These arguments left only combustion gas recirculation effects as the most plausible way for the heat exchanger to influence NO production in the flame zone. It was reasoned that the strength and direction of the burner air jet, together with density gradients set up by the water-cooled heat exchanger, induced a vertical recirculation pattern that was upward at the back combustor wall (opposite the burner) and downward at the front wall. An experiment was devised, therefore, to see if the beneficial effects of the water-cooled coil could be simulated by forcing a flow of cooled, recirculated combustion gases toward the burner side of the flame zone. For convenience in cooling and pumping, flue gas was used as the recirculant, and it was put back into the combustor through an existing peephole, as shown in Fig. 17. Also shown are two additional gas-sampling locations ("B" and "C") that were installed to attempt to track the formation of NO within the furnace. Not shown in this schematic is a 0.11 m diameter pyrex window added to the top of the central combustor cylinder to observe the flame during flue gas recirculation.

Data resulting from experiments with this apparatus are listed in Table C-3. The "Recir. Ratio" (listed in %) is defined as the ratio of recirculated stoichiometric burned gas to total "unburned" air. This definition deducts excess air from the recirculated combustion gases and combines it with fresh air injected by the burner so that the recirculation ratio is corrected for changes in overall stoichiometric ratio conditions. Very large amounts of recirculated burned flue gases (40 to 50%) were required to lower the flue gas NO concentration below 0.6 g/kg (Runs 304 to 329). An additional small water-cooling coil was added to the FGR circuit to further cool the recirculated gases (Runs 340 to 346). The resultant 50 C reduction in temperature of the recirculated gas was found to achieve an insignificant (~-0.04 g NO/kg) further reduction in NO emissions. It was believed to be unlikely that the large amounts of flue gas recirculation that were required in these tests to reduce the NO concentrations significantly could be induced by gravitational effects on density gradients caused by the presence of the water-cooled coil. It was thought a cold-coil-induced CGR flow pattern in the opposite direction of rotation, i.e., rising on the front wall instead of the rear wall, might be possible. However, no further mechanically pumped FGR experiments were conducted.

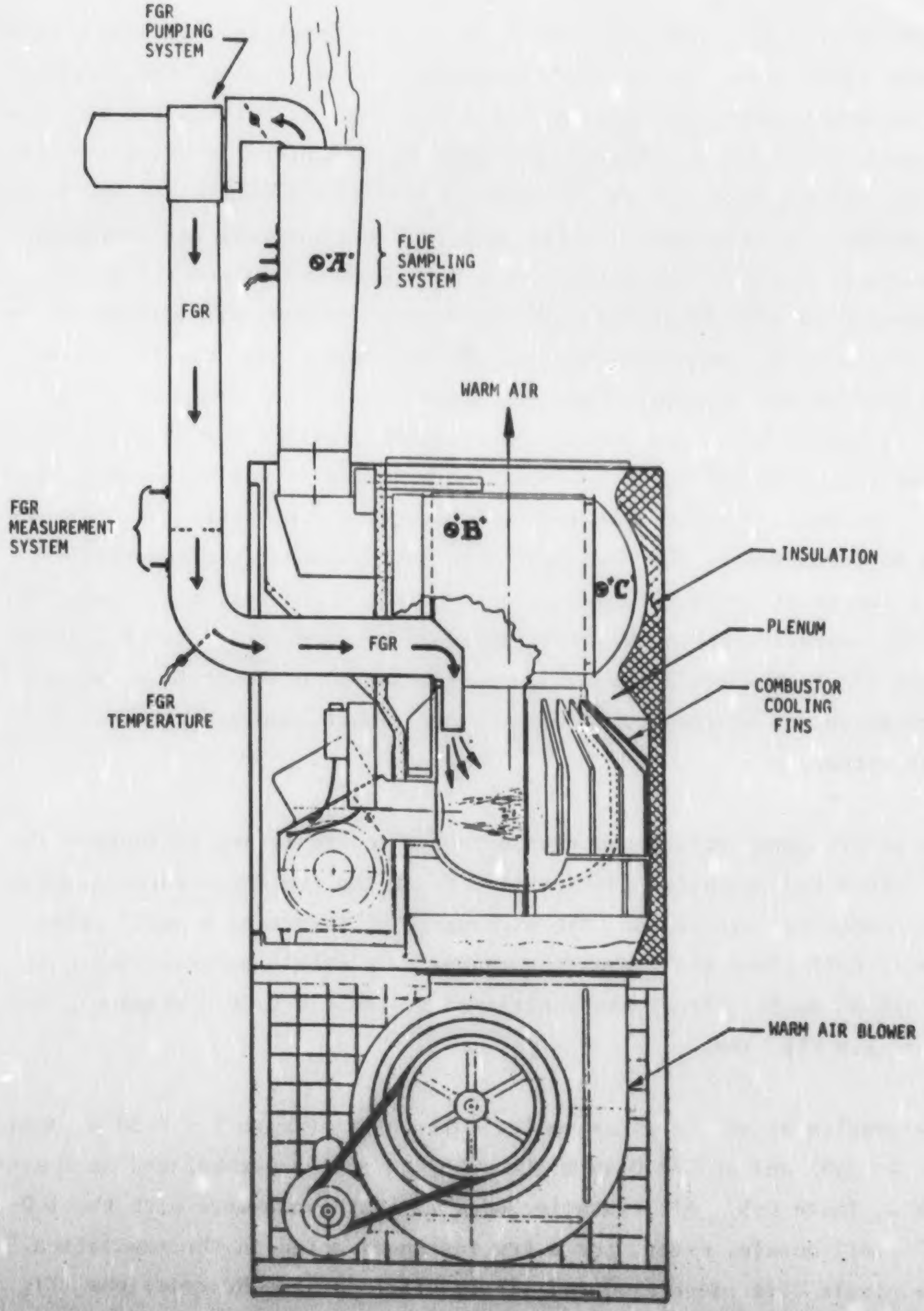
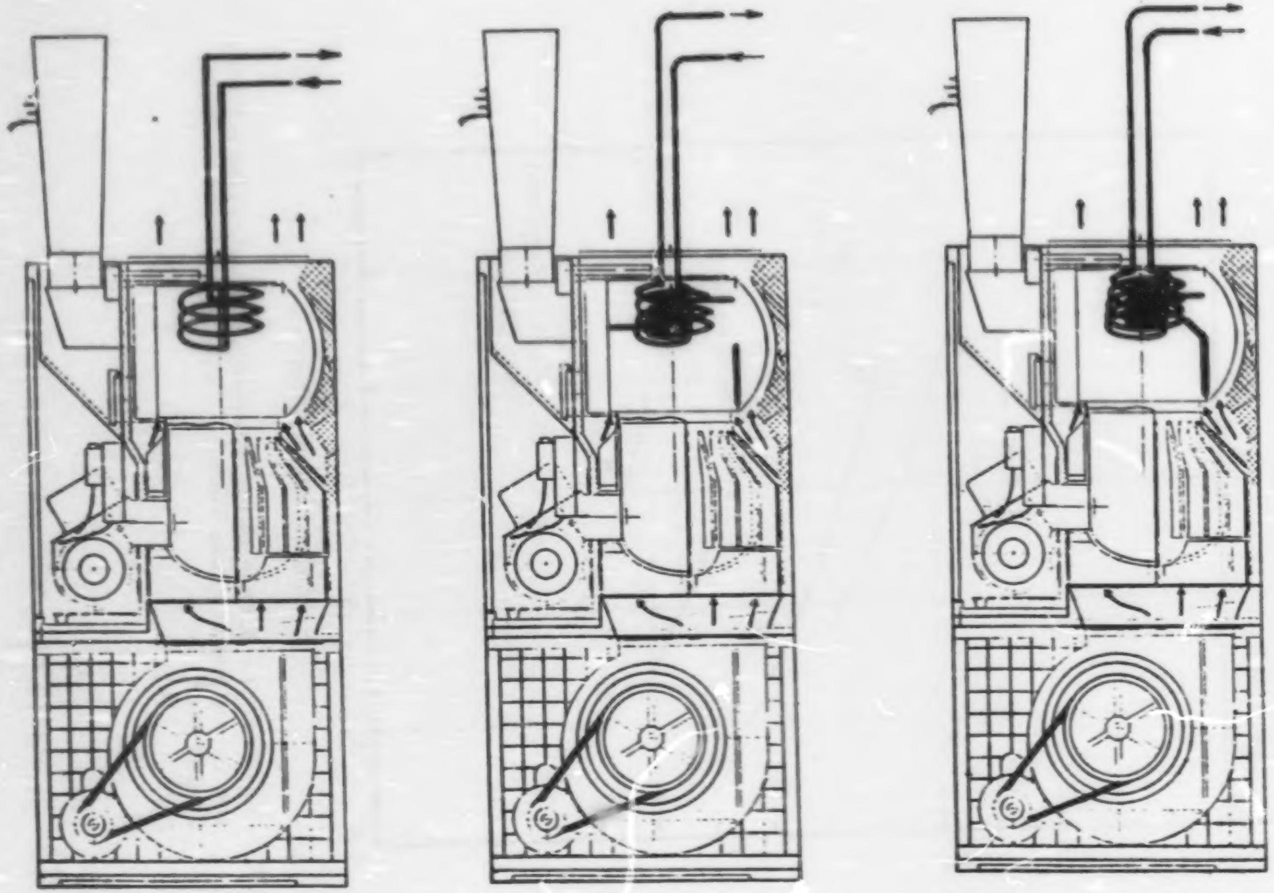

FGR PUMPING SYSTEM
FLUE SAMPLING SYSTEM
'A'
FGR
WARM AIR
FGR MEASUREMENT SYSTEM
'B'
'C'
INSULATION
PLENUM
FGR
COMBUSTOR COOLING FINS
FGR TEMPERATURE
WARM AIR BLOWER

Figure 17. Schematic of the Prototype Optimum Furnace With an Experimental Flue Gas Recirculation System

A series of experiments was conducted in an attempt to determine whether NO was being formed in the heat exchanger. This involved the addition of two sampling probes, labeled B and C in Fig. 17, within the heat exchanger. Location B extracted gas samples from directly above the combustor and location C from the entry to the second section of the heat exchanger. A comparison of flue samplings and internal gas samplings (locations C and B) may be made from data gathered in several tests (Runs 330 to 339, Table C-4). The data revealed some differences in the completeness of combustion (CO and UHC emissions), but the NO concentrations showed no significant differences among the flue and the internal sample locations. This indicated that all of the NO was actually being formed in the firebox combustion zone (as it was believed it must be). However, the influences which downstream components can exert upon the NO formation in the combustor have been demonstrated repeatedly. This series of tests gave more credence to the combustion gas recirculation concept than to the distributed-quench concept of NO production, since the latter would have shown increasing NO concentrations as the combustion gas progressed through a more slowly quenching, air-cooled heat exchanger.

The experimental effort was then directed toward trying to improve the emissions and operating characteristics of the prototype furnace, with its commercial air-cooled heat exchanger, by inserting a small water-cooled coil above the combustion chamber. A single water-cooled coil (0.116 m^2 surface area) was positioned at the L = 0.50 m chamber position (see Fig. 18a).

The results of single water-cooled coil tests at both L = 0.50 m (Runs 347 to 356) and at L = 0.40 m (Runs 357 to 366) are tabulated in Appendix C, Table C-5. All tests included in that table were with the 1.0-60°-A oil nozzle, except for a few instances noted in the tabulation. The single coil alone produced little effect on the NO emissions (Fig. 19), so a baffle was added to the entrance to the rear manifold which forced the combustion gases toward the coil before they could enter that



(a) Single Coil

(b) Double Coil &
Rear Exit Baffle

(c) Double Coil & Extended
Rear Exit Baffle

Figure 18. Prototype Optimum Furnace With Various Configurations of Internal Water-Cooled Coil and Baffles

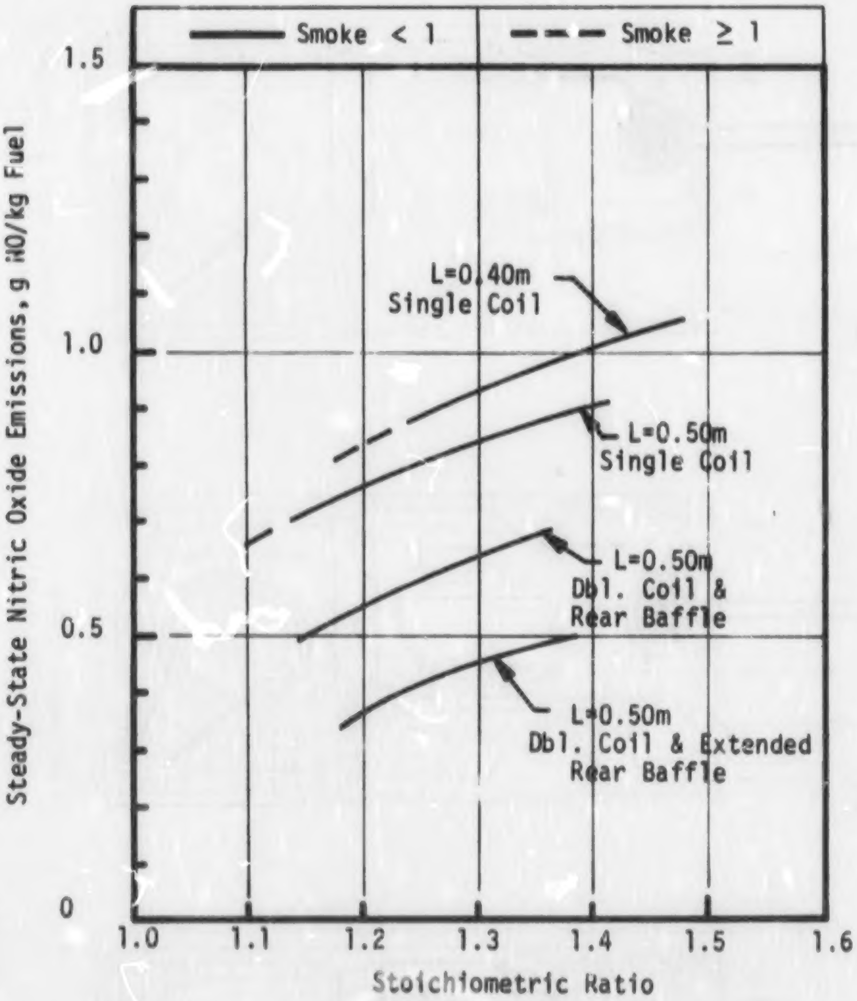


Figure 19. Steady-State Nitric Oxide Emission Concentrations From the Prototype Optimum Furnace With an Additional Water-Cooled Coil

manifold. The results from this configuration again showed very little effect upon the NO concentrations (Runs 367 to 373). Another coil was then added within the original coil to increase the cooling surface area to 0.18 m^2 (Fig. 18b). This configuration did have an effect upon the combustion process (Runs 374 to 401). The NO concentrations dropped from 0.80 to 0.55 g/kg, but the CO concentrations showed some sensitivity to this configuration.

To investigate whether the influence on the NO emissions stemmed from combustion gas recirculation or radiation cooling of the flame zone, an extension was added to the rear baffle (Fig. 18c). This extension was canted to 45 degrees from vertical to induce a downward flow of recirculant gases on the back wall. This is opposite to the direction of the recirculation rotation that was imposed mechanically in the FGR experiments described earlier in this section. A total of 24 firings (Runs 402 to 425, Table C-5) was made with this extended rear baffle. The results with water and steam cooling showed further substantial reductions in NO emissions, adding support to the vertical combustion gas recirculation hypothesis. The tests with no cooling coils (Runs 420 to 425) above the combustor showed no reduction in NO emissions, implying that the recirculation is not a simple case of gas dynamics, but involves the cooling coils to cool some of the combustion gases (i.e., increase gas density) to promote recirculation by gravitational effects.

To isolate this cooling-to-induce-recirculation from the simple rapid-quench suppression of NO production, the existing double-cooling coil was removed and a smaller cooling coil was installed. This new cooling coil was only 0.05 m outside diameter, spiraled horizontally, and was tucked under the overhang of the 45-degree extended baffle lip. This coil was exposed to only a fraction of the combustion gas stream, and it would be unlikely to influence the NO kinetics of the bulk of the gases leaving the combustor. The fraction of the gas that it cooled was expected (by the positioning of the coil under the baffle lip) to be limited to the vertically recirculated combustion gases. The influence of this small, well-positioned cooling coil on the combustion process was significant (Runs 426 to 433, Table C-5), showing a reduction in NO concentration from about 0.85 g/kg (no coil, Runs 420 to 425) to about 0.65 g/kg. The CO concentrations showed a marked increase, an additional indication that the flame zone was influenced by the small coil.

Along with the increase in CO emissions came as associated increase in combustion oscillation. As a simple exploratory measure, a 0.15 m diameter by 0.81 m long closed-end pipe was extended above the central combustor/heat exchanger cylinder (0.25 m diameter by 0.69 m long) to change the natural resonant frequencies of the system. This improved the operational characteristics of the system somewhat (less combustion oscillations) and, surprisingly, resulted in a further reduction (-0.15 g/kg) in NO emissions (Runs 434 to 443, Table C-5). It was thought that noisy combustion may cause transient departures of the flame zone from its near-optimum mixing and burning conditions for producing minimum NO, and that reducing combustion oscillations restored the flame to the optimum combustion conditions. Evidence of this effect will be seen again in a later discussion of the final optimization modifications to the prototype furnace system.

The question as to whether the horizontal entry to the heat exchanger's rear manifold could be contributing to the differences in operational characteristics from those of the vertical combustor exhaust configuration had not been fully resolved. Therefore, a set of experiments was carried out with another heat exchanger configurational change. The air-cooled heat exchanger was elevated by placing a 0.35 m long spool of 0.25 m diameter pipe between it and the combustion chamber. The double-coil exchanger was centered near the top of the spool. Combustion gases were exhausted vertically upward, through a 0.18 m diameter port at the top of the spool, after passing over the coil. A 90-degree elbow then turned the flow into the air-cooled heat exchanger's rear manifold. This configuration was tested with water coolant in the coils

(Runs 444 to 447, Table C-5), and with reduced water flowrate to produce steam (Runs 448 to 451). Full water cooling produced exceptionally low NO levels, but high CO and noisy combustion prevented operation at acceptably low stoichiometric ratios. Reduced cooling of the coils lowered the CO emissions somewhat, but also sharply increased the NO production.

An attempt was made to reproduce the results obtained earlier with the internal water-cooled coil configuration (Fig. 18c) using a low pressure, air-cooled device. The low pressure requirement was added to eliminate the need for an air compressor system within the furnace if the results led to any working prototype. This air-cooled coil design (Fig. 20) differed from the water-cooled coil design in that the spiraling tubes formed eight parallel paths, i.e., lower pressure drop. The air coolant was initially blown by a burner fan and was later augmented by using two such fans in series. Air entered the top of the combustor canister through a 0.051 m outside-diameter steel tube, which extended down the center to the bottom of the coil heat exchanger assembly, then was manifolded out radially into four 0.013 m outside-diameter and four 0.006 m outside-diameter copper tubes that spiraled back to near the point of entry. The diameter of the outer coils was 0.146 m, and the coil assembly was shrouded by a 0.152 m outside diameter by 0.229 m long stainless-steel, sheet-metal cylinder. The shroud was added to induce separate flow paths: external for upflowing hot gases and internal for downflowing cooled, recirculant gases. Test results are presented in Table C-6, Runs 491 to 511, with minor changes in geometry noted on the tabulation. The results showed that no benefits were realized with the installation of this air-cooled coil. The minimum of 0.62 g/kg (CO $\leq$ 1.0 g/kg, UHC $\leq$ 0.1 g/kg, no smoke) at S.R. = 1.14 obtained with the basic prototype was not surpassed by any of the four configurations of the low pressure, air-cooled coil system.

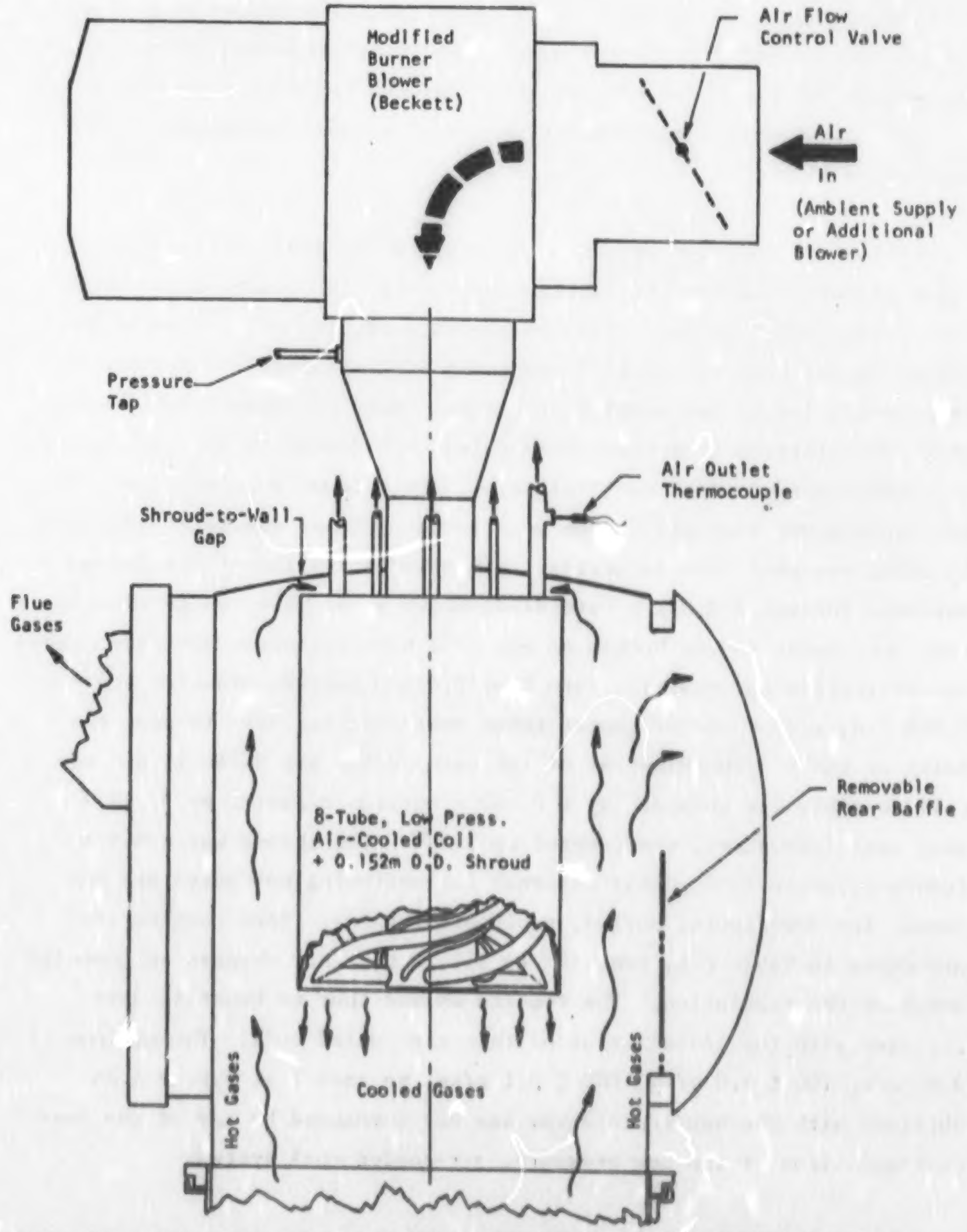

Modified
Burner
Blower
(Beckett)
Air Flow
Control Valve
Air
In
(Ambient Supply
or Additional
Blower)
Pressure
Tap
Air Outlet
Thermocouple
Shroud-to-Wall
Gap
Flue
Gases
Removable
Rear Baffle
8-Tube, Low Press.
Air-Cooled Coil
+ 0.152m O.D. Shroud
Hot Gases
Cooled Gases
Hot Gases

Figure 20. Schematic of the 8-Tube, Low-Pressure, Air-Cooled, Supplemental Heat-Exchanger Installation in the Prototype Optimum Furnace

DISCUSSION

The prototype optimum furnace, as designed and with the original optimum burner from the Ref. 1 studies, fell somewhat short of achieving its design goals. Of the many modifications which were tested experimentally, that which came closest to satisfying the goals was retaining the prototype furnace design unchanged and refining the burner design. That system met all of the goals except the one for NO_x emissions: at the nominal design point, cycle-averaged NO was below 0.65 g NO/kg fuel burned, as compared with the target level of 0.5 g/kg.

NO_x emissions under that target were attained by some selected modified configurations. Invariably, however, those configurations either exhibited undesirable attributes concerning some other pollutant or involved a "hybrid" heat transfer system (using both air and water coolants) or both. It may be inferred that a hydronic boiler embodying the low-emission burner and combustion chamber design criteria could readily meet all the design goals. A warm-air furnace design with a portion of its combustion gas cooling accomplished by a water-cooled heat exchanger, on the other hand, would be at a competitive disadvantage because of the additional complexity and cost of providing simultaneously for combustion gas-to-air, combustion gas-to-water, and water-to-air heat transfer. As a result, the remainder of this discussion is concerned predominantly with the air-cooled prototype furnace with the refined design optimum burner. In fact, from this point forward, the phrase "optimum burner" will be applied to that refined design and the phrase "prototype optimum furnace" will be used for the prototype unit in which that burner was tested.

Comparison With Other Residential Furnaces

Pollutant Emissions - Flue gas concentrations of NO are plotted versus stoichiometric ratio in Fig. 21. A shaded region near the middle of the graph indicates that a large majority of existing residential oil

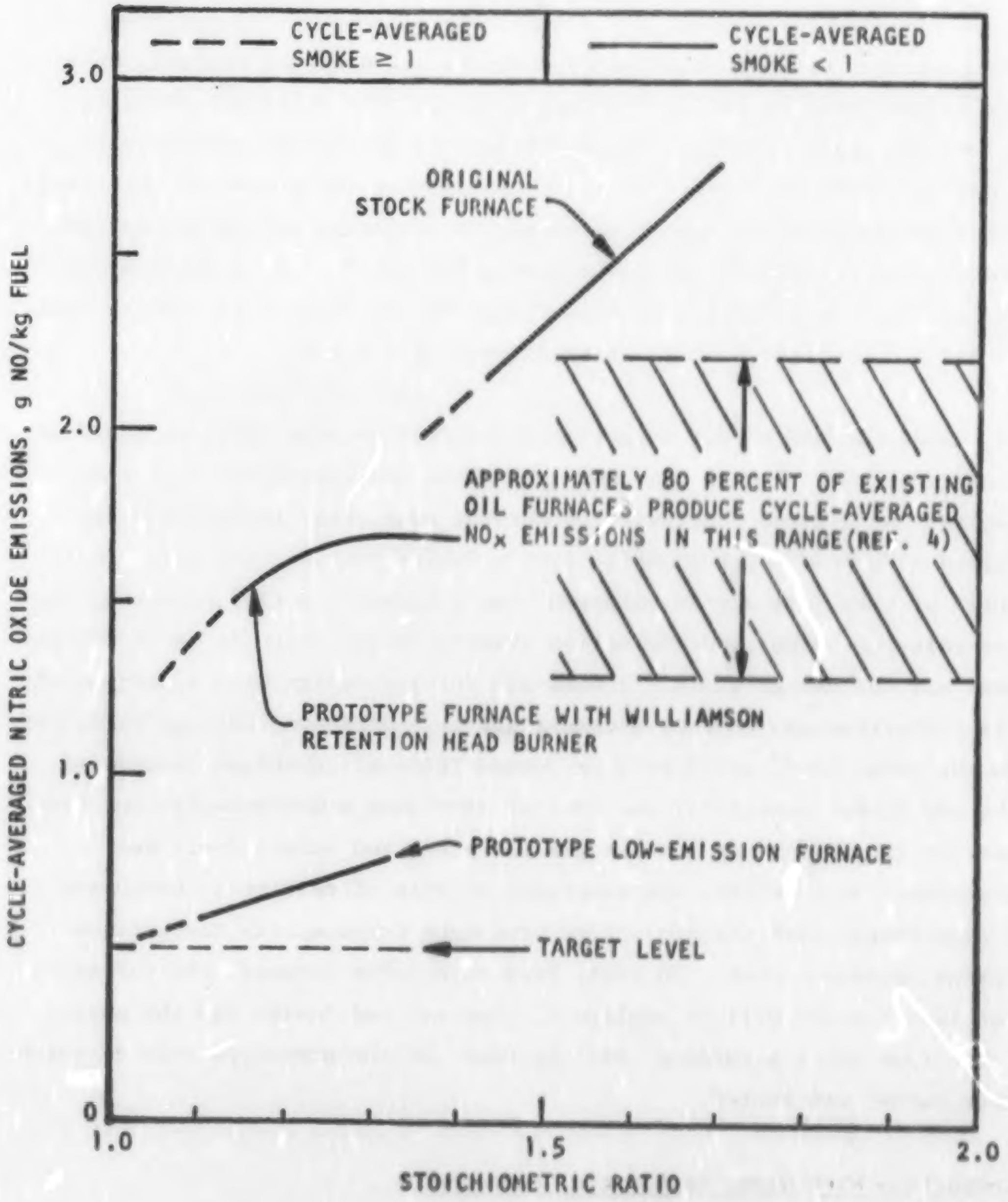


Figure 21. Comparison of Cycle-Averaged NO Emissions From the Prototype Optimum Furnace and Other Oil Furnaces

furnaces release between 1.3 and 2.2 g NO/kg fuel burned. An overall average level of 1.8 g NO/kg fuel may be used for evaluating the potential impact of applying candidate NO_x reduction techniques (derived from Ref. 4).

Measured NO emissions from the stock furnace fell on the high side and above that typical range; at a nominal 50% excess air operating point, it produced 2.2 g NO/kg fuel burned. Measured NO emissions from the prototype optimum furnace were much lower. Tuned to the intended normal operating condition with only 15% excess air, the unit produced 0.63 g NO/kg. That corresponds to reductions of about 72 and 65%, respectively, from NO_x emissions produced by the stock furnace (at its nominal operating point) and by the average estimated for all existing installed units.

The Williamson flame retention burner, that was known to operate well in the stoichiometric ratio range of interest, was fired in a series of tests in the prototype optimum furnace. The data (Fig. 21) showed that this combination produced intermediate-level NO_x emissions (1.5 to 1.7 g NO/kg fuel). This clearly shows that the optimized, finned, air-cooled firebox is beneficial in its own right, since NO_x was reduced by about 27% from the stock furnace's emission level, but is most effective when combined with the optimum burner.

Carbonaceous emissions from the prototype furnace unit also were acceptably low at its nominal conditions, as indicated by the lower-than-No. 1 smoke. A comparison of values in Table 2 shows that CO and hydrocarbon emission levels from the prototype furnace were somewhat higher than those measured for the stock furnace, but were quite comparable with averaged tuned values measured in the field survey of Ref. 4.

Table 2. COMPARISON OF FURNACE OPERATING CONDITIONS AND CYCLE-AVERAGED POLLUTANT EMISSIONS

	Tuned Averages From Ref. 4	Stock Lennox Furnace	Prototype Optimum Furnace
Stoichiometric Ratio	1.85	1.50	1.15
Carbon Monoxide, g CO/kg Fuel	0.6	0.27	0.55
Unburned Hydrocarbons, g UHC/kg Fuel	0.07	0.015	0.055
Smoke, Bacharach Number	1.3	0	0
Nitric Oxide, g NO/kg Fuel	1.8	2.2	0.63

Efficiencies - Steady-state efficiencies measured for the prototype optimum furnace are compared with those for its stock predecessor and other residential units in Fig. 22 by superimposing the measured data on the ANSI efficiency decrement plot of Fig. 8. Based on data from a number of sources, it is estimated that a large majority (perhaps as high as 80%) of existing installed residential oil heating systems operate in the shaded zone in the right-central portion of Fig. 22. Older existing units tend to perform toward the upper and right-hand regions of that zone, while newer equipment tends to congregate in the lower and left-hand regions. Obviously, substantial numbers of units also operate outside that shaded zone, and they are distributed around it on all four sides. The average behavior of all United States oil-fueled heating systems probably lies in the central crosshatched region of that zone, with net flue temperatures in the neighborhood of 280 C (500 F), CO_2 concentrations of around 8%, and estimated average steady-state efficiencies between 72 and 75%.

The performance curve for the original stock furnace fell well below (i.e., at higher efficiencies) the shaded band representative of existing installed residential heating units. The stock unit could be tuned to a moderately low 50% excess air nominal operating condition where its

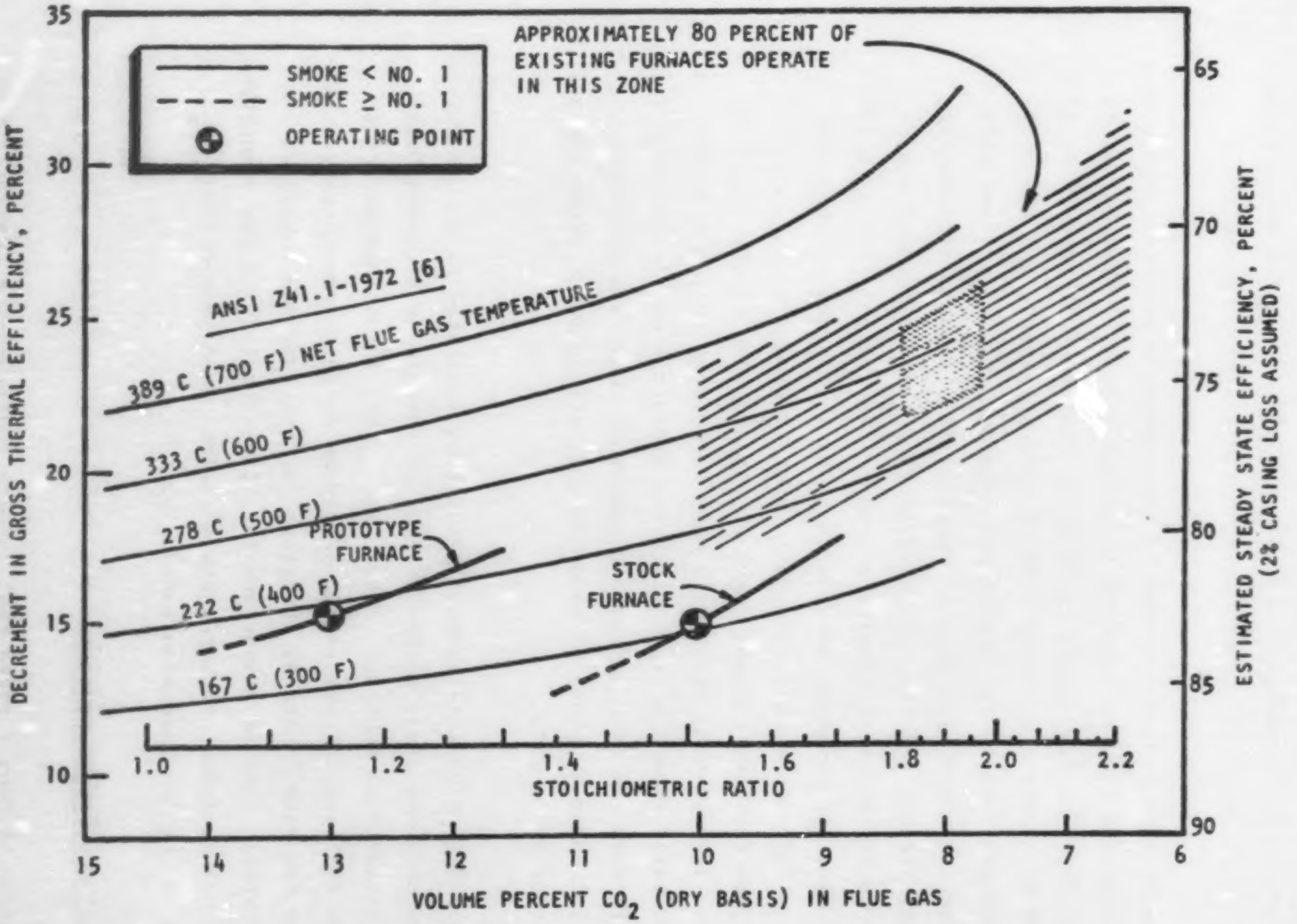


Figure 22. Comparison of Steady-State Thermal Efficiencies of the Prototype Optimum Furnace and Other Oil Furnaces

net flue gas temperature averaged only 180 C (325 F). The resultant steady-state gross thermal efficiency was 82.5% (i.e., the stock furnace was among the higher-performing units in the existing population).

Thermal efficiency levels achieved by the prototype optimum furnace were qualitatively the same as those of the stock furnace. However, as is evident in Fig. 22, flue gases leaving the optimum unit were 40 to 55 C (75 to 100 F) hotter than those from the stock furnace, and the efficiency decrement due to the higher net flue gas temperature was offset by operating the prototype unit at substantially lower stoichiometric ratio. The thermal behavior of the prototype optimum furnace was surprising because the stock furnace's compact heat exchanger was retained intact and was supplemented substantially by the finned firebox heat exchanger. This apparently anomolous behavior probably was caused by warm-air jets from the firebox region bypassing some of the main heat exchanger. It should be relatively easy to correct this condition. It can be estimated from Fig. 22 that if the prototype unit's net flue gas temperature were the same as that of the stock unit, its efficiency would be increased by about 2% to an overall steady-state gross thermal efficiency of 84 to 85%.

The 82 to 83% steady-state thermal efficiency exhibited by the prototype optimum furnace was close to the maximum achievable in noncondensing flue gas residential systems. Taken alone, this is not unique, since comparable efficiencies are attained by some current commercially available units (as exemplified by the stock furnace that was converted into the prototype). What is unique and important about it is the demonstration that near-maximum, steady-state efficiency and near-minimum NO_x emissions can be obtained simultaneously.

Operational and Design Aspects

The prototype optimum furnace test results confirmed the feasibility of applying the several newly developed, low-emission, oil burner and firebox design criteria to residential space-heating equipment. The experimental prototype unit came very close to satisfying all of the pollutant emission and efficiency objectives for which it was designed. Operationally, its behavior was quite comparable with current commercially availabe furnaces. The 500-hour-duration test, equivalent to about one-tenth of an average heating season, indicated that the unit might serve through an entire no-maintenance winter heating season without exhibiting appreciable shifts in operating conditions or pollutant emission levels.

As delineated in the statement of the design criteria, even greater NO_x reductions may be achieved by adopting tunnel-fired burner orientations and/or water-cooled combustor walls in furnace designs. Some beneficial reductions in NO_x emissions (up to about 25%) were achieved by invoking either the burner design criteria alone or the firebox design alone, but the NO_x emission goal could be approached only by using both sets of criteria in combination. Furthermore, there was no assurance that either set of criteria alone could minimize the excess combustion air requirement.

The decrement between steady-state and cycle-averaged efficiencies of the prototype optimum furnace was smaller than the uncertainty in measuring the latter value in the outdoor laboratory. As a result, quantitative assessments were not obtained for features included in the prototype unit to cut down on cyclical heat losses. Elimination of draft air heat losses was the most important of these. Although it was anticipated that some burner or electrical control overheating during standby might result from sealing the burner vestibule, absolutely no indication of any problem was observed during the 500-hour test. Presumably, the metallic firebox was not hot enough to cause a radiation problem, and conduction was acceptably low.

In view of its demonstrated steady-state efficiency and the apparent effectiveness of its features for reducing standby heat losses, the prototype optimum furnace should achieve 75% or higher cycle-averaged efficiency. This is perhaps 13% higher than the estimated mean season-averaged performance of existing United States residential oil heat sources that it might replace. Such replacement would be attended by an average of about 17% reduction in fuel consumption. This estimated lower fuel consumption may be combined with the 65% reduction in NO_x (as normalized by the mass of fuel burned) to calculate the total effect on mass emissions. The result is that, if a prototype optimum furnace replaced an "average" existing unit and satisfied the same thermal demand, the mass of NO emitted would be reduced by 71%.

Reductions in fuel consumption brought about by the sealed barometric and combustion air supply system have not been included in the foregoing discussion. This type of sealed air system has been shown in laboratory testing (Ref. 5) to reduce residential heating oil consumption by a minimum of 5% and up to 15%. An approximate average value for its potential effect on optimum furnace fuel consumption is probably a little above the minimum, say 8%. Adding this to the estimated 17% due to higher unit efficiency yields an average anticipated fuel savings of 25%. Corresponding to that is an estimated 74% overall reduction in the mass of NO_x emissions.

SECTION V

INTEGRATED SYSTEM DESIGN

It should be clear, from the foregoing description and discussion of test results, that the prototype furnace closely approached an optimum design with respect to air pollutant emissions and performance. Additional attention was given to some economic considerations, such as unit weight, materials costs, and fabrication methods, and design details were derived for a candidate, integrated, low-emission, warm-air furnace. The integrated system design is described in this section in terms of those design features which differ from the prototype furnace described in Section IV.

A cutaway perspective drawing of the integrated furnace is shown in Fig. 23. This is supplemented by several views from an assembly drawing in Fig. 24. Components which differ from the prototype furnace design (Fig. 2) are the burner assembly, the finned combustor, the baffles in the warm-air flow passage, and the burner vestibule closure panels. Details are described in the following subsections.

BURNER ASSEMBLY

The basic burner assembly in the integrated furnace design is the optimum burner as it was finally tested in the prototype furnace. That assembly included the standby draft control device, the quiet stator on the fan discharge, a static disc in the blast tube, and the research optimum head. Two changes have been incorporated for the integrated furnace design, as follows:

Combustion Air Control Device

A potential problem was cited in Ref. 3 in satisfying applicable safety standards with the prototype standby draft control unit that relies upon burner fan suction to open a flap valve held closed by gravity. Although

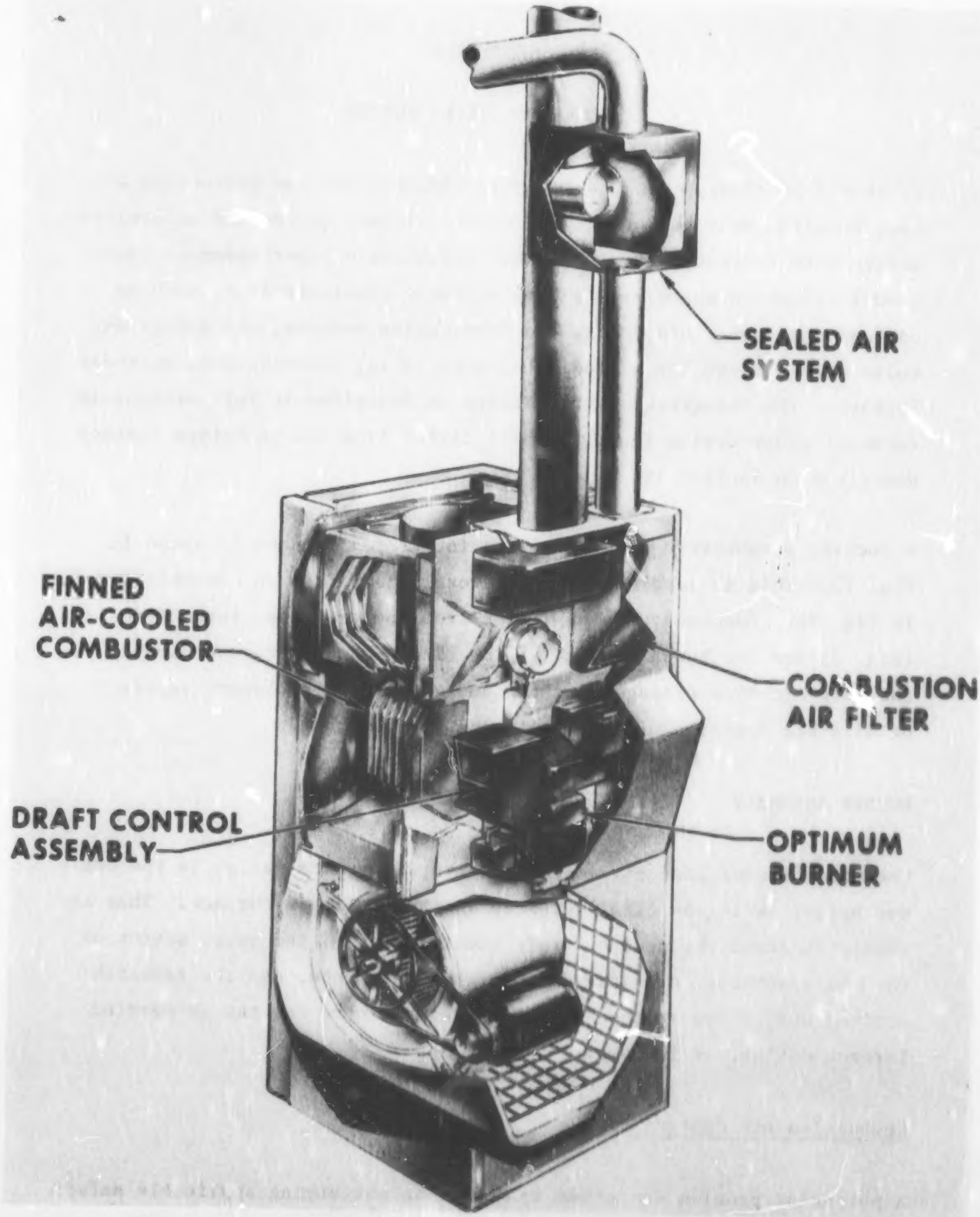
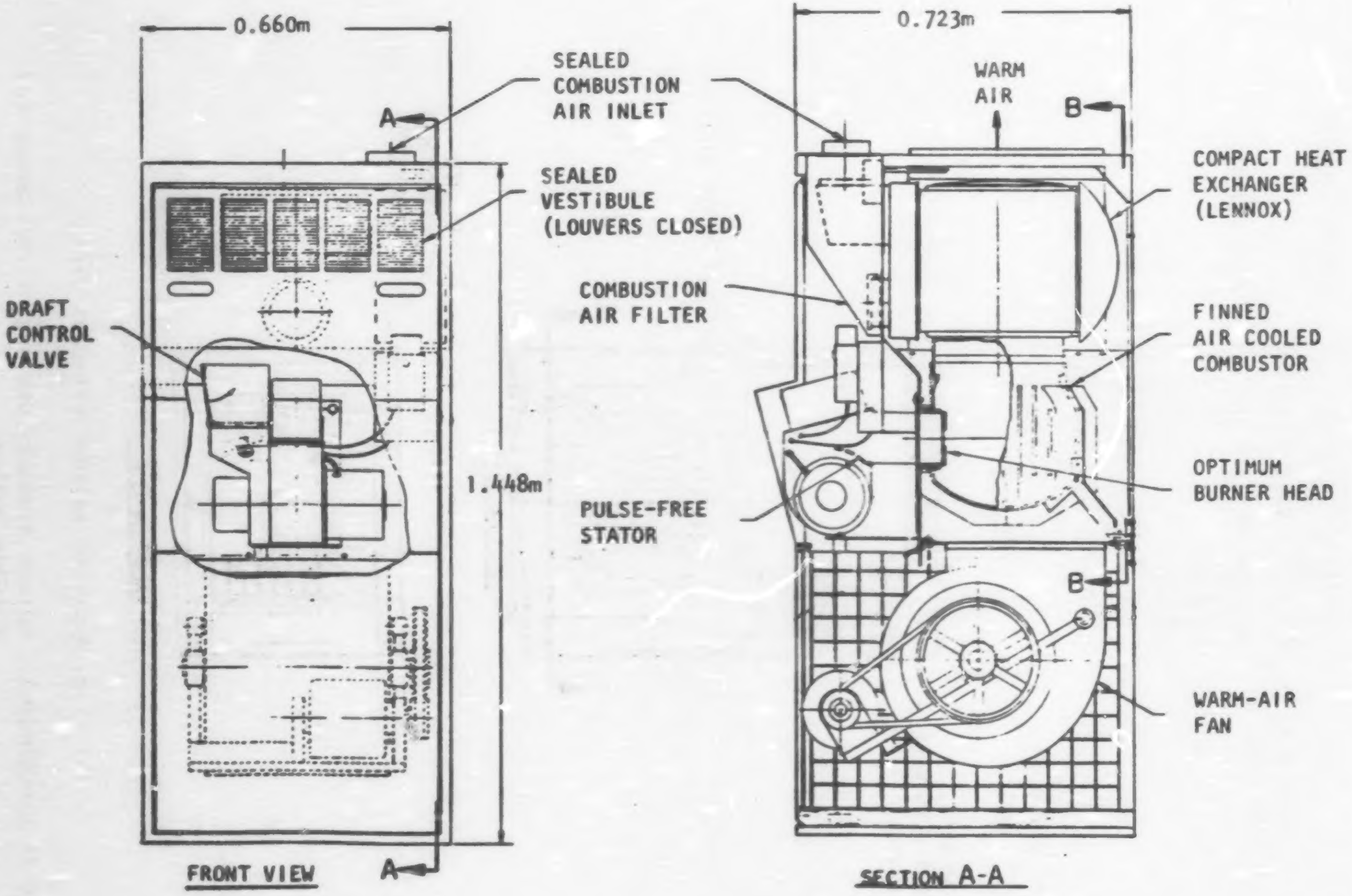
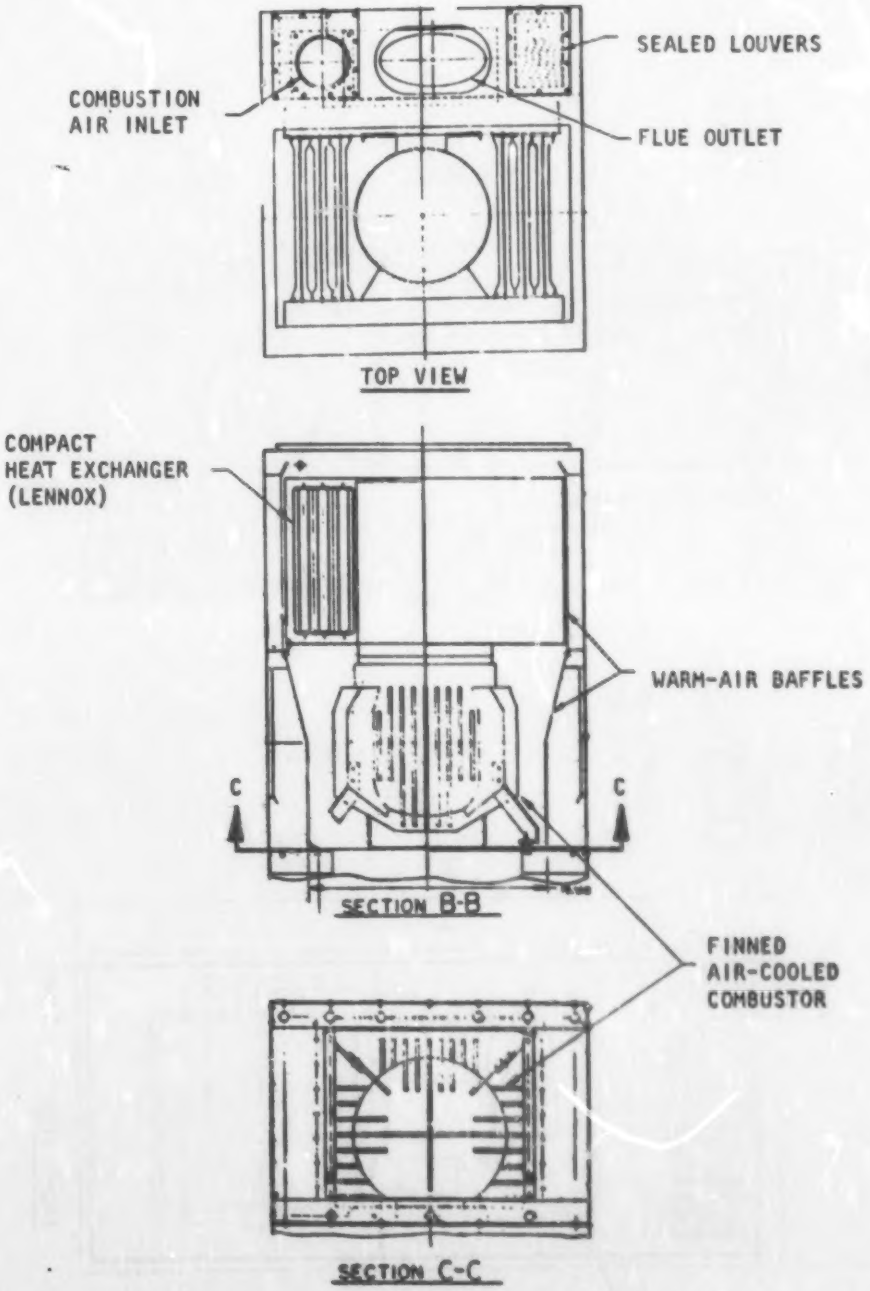

SEALED AIR SYSTEM
FINNED AIR-COOLED COMBUSTOR
COMBUSTION AIR FILTER
DRAFT CONTROL ASSEMBLY
OPTIMUM BURNER

Figure 23. Cutaway Perspective Drawing of Integrated, Low-Emission, Warm-Air Furnace Design


0.660m
SEALED COMBUSTION AIR INLET
A
SEALED VESTiBULE (LOUVERS CLOSED)
DRAFT CONTROL VALVE
1.448m
A
FRONT VIEW
0.723m
WARM AIR
B
COMPACT HEAT EXCHANGER (LENNOX)
COMBUSTION AIR FILTER
FINNED AIR COOLED COMBUSTOR
OPTIMUM BURNER HEAD
PULSE-FREE STATOR
B
WARM-AIR FAN
SECTION A-A

(a) Front and Side Sectional Views

Figure 24. Layout Assembly Drawing of the Integrated Furnace Design


SEALED LOUVERS
COMBUSTION AIR INLET
FLUE OUTLET
TOP VIEW
COMPACT HEAT EXCHANGER (LENNOX)
WARM-AIR BAFFLES
C
C
SECTION B-B
FINNED AIR-COOLED COMBUSTOR
SECTION C-C

(b) Top, Rear and Bottom Sectional Views

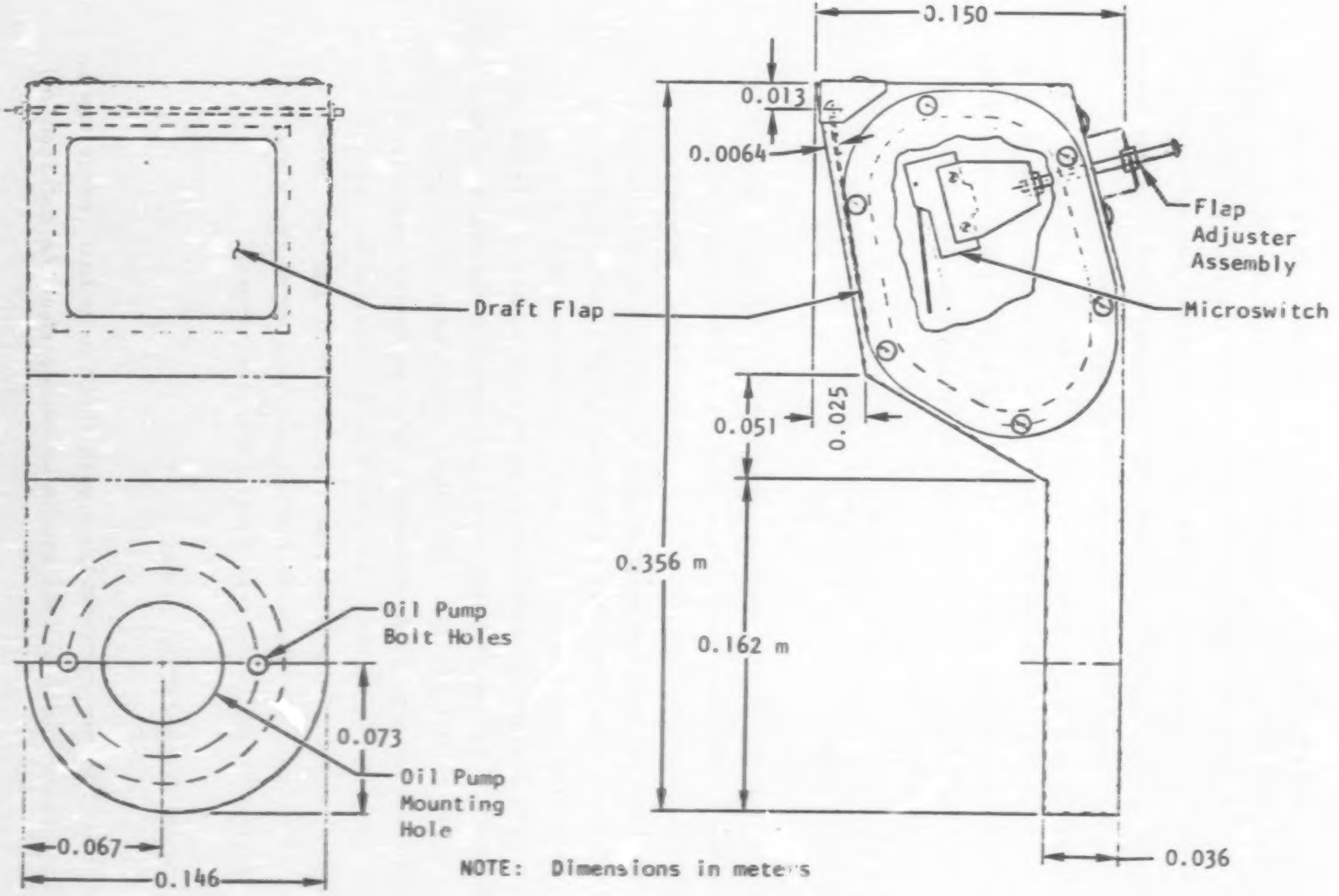
Figure 24 (concluded). Layout Assembly Drawing of the Integrated Furnace Design this device functioned well in the laboratory testing, it was anticipated that an interlock with a solenoid shutoff valve in the fuel supply line, to ensure that the flap is open before fuel can flow to the burner, might be required to satisfy the Underwriters' Laboratories standard (Ref. 8). Therefore, a combustion air control device has been designed in which the shutoff flap, upon opening, closes a normally open microswitch. This is illustrated in Fig. 25. The microswitch is mounted on the end of a threaded rod which also provides a positive stop for the air flap. Adjustment of the rod's insertion depth controls the distance that the flap can open, and so can be used to control the combustion air flowrate.

Optimum Burner Head

The research version of the optimum burner head, fabricated by machining and welding stainless-steel plate (Ref. 1), was utilized throughout the testing of the prototype furnace. Less expensive, commercially practicable head fabrication methods were considered in the study reported in Ref. 2. The preferred method was found to be stamping and folding heads from stainless-steel sheet. Prototype heads made to simulate those which might be made commercially were tested and found to reproduce quite well the performance of the research optimum head and, potentially, to be durable and long-lived. Therefore, the stamped sheet metal optimum head is incorporated into the burner assembly for the integrated furnace design. This head is illustrated in Fig. 26. (reproduced from Ref. 2) as a composite plan view of the flat sheet-metal stamping and a rear view of the optimum head after the six swirl vanes have been folded twice into their final positions.

FINNED AIR-COOLED COMBUSTION CHAMBER

The combustion chamber was redesigned with three principal objectives in mind: (1) fabrication by a less expensive method than the machined and


0.150
0.013
0.0064
Flap
Adjuster
Assembly
Draft Flap
Microswitch
0.051
0.025
0.356 m
0.162 m
Oil Pump
Bolt Holes
0.073
Oil Pump
Mounting
Hole
0.067
0.146
0.036

NOTE: Dimensions in meters

Figure 25. Drawing of the Combustion Air Control Device

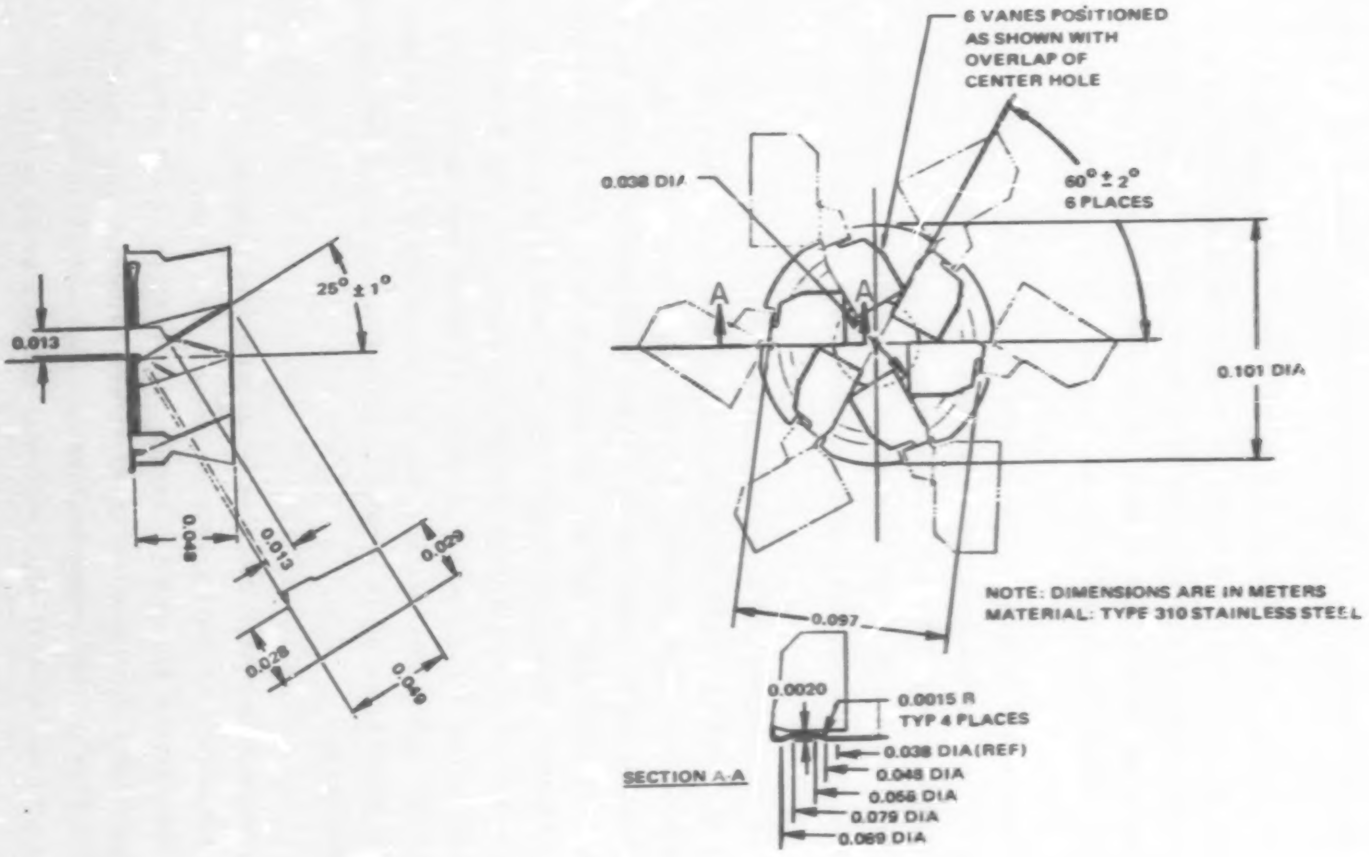

6 VANES POSITIONED AS SHOWN WITH OVERLAP OF CENTER HOLE
0.038 DIA
60° ± 2° 6 PLACES
A
A
0.101 DIA
0.097
NOTE: DIMENSIONS ARE IN METERS
MATERIAL: TYPE 310 STAINLESS STEEL
0.0020
0.0015 R TYP 4 PLACES
0.038 DIA(REF)
0.048 DIA
0.056 DIA
0.079 DIA
0.089 DIA
SECTION A-A
25° ± 1°
0.013
0.048
0.013
0.029
0.028
0.049

Figure 26. Stamped and Folded Sheet Stainless-Steel Optimum Head

welded construction of the prototype unit, (2) reduction of weight, and (3) provision for joining the firebox to the heat exchanger by welding.

The new firebox design is illustrated in Fig. 27. Metal casting was selected as the best and cheapest way of making the relatively complicated finned combustor assembly. The fins were changed from a predominantly radial orientation to a predominantly parallel orientation to simplify and reduce the number of pieces needed for a casting mold.

The design provides a rolled carbon-steel ring, to be fitted in the mold and integrated into the casting, that matches and is to be welded to the central cylinder of the fabricated sheet metal heat exchanger. The ring is perforated with a number of countersunk holes that, when filled with casting metal, are intended to ensure permanency of the bond.

The design of the cast-metal firebox provides typical wall and fin thicknesses of 0.0063 m (0.25 inch), which is close to the minimum for reliable casting of a unit of this size and complexity. The reduced metal thickness helps to lower firebox mass. Together with some shortening of the redistributed fins and the elimination of the bolted heat exchanger attachment rings, the total mass has been reduced from the 61.2 kg (135 lbm) of the prototype finned combustor into the 27 to 30 kg (60 to 66 lbm) range for a cast-iron unit. Although that would be on the order of 50% reduction from the mass of the existing prototype unit, it can hardly be said to be in economic competition with current commercial construction.

Further appreciable weight reductions would probably require either reducing the dimensions of the firebox or using a less dense construction material. The firebox dimensions, having been derived from burner emission and performance optimization studies, are no longer considered to be optional variables, but a material change may well be acceptable. Cast aluminum is the most likely candidate; its use would result in a

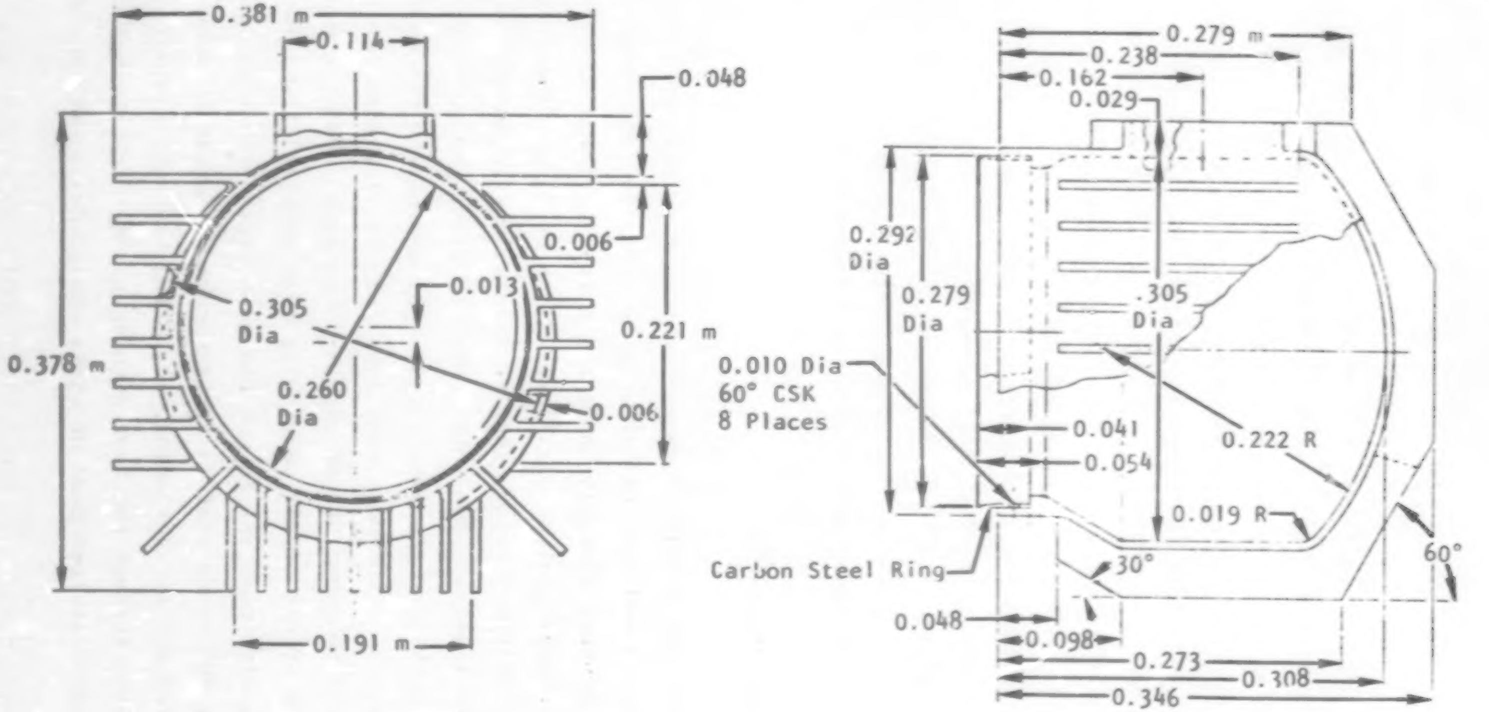

0.381 m
0.114
0.048
0.006
0.013
0.305
Dia
0.221 m
0.378 m
0.260
Dia
0.006
0.191 m
0.279 m
0.238
0.162
0.029
0.292
Dia
0.279
Dia
.305
Dia
0.010 Dia
60° CSK
8 Places
0.041
0.054
0.222 R
0.019 R
Carbon Steel Ring
30°
60°
0.048
0.098
0.273
0.308
0.346

Figure 27. Cast-Metal Finned, Air-Cooled Combustion Chamber

firebox weight in the 10 to 12 kg (22 to 26 lbm) range. Before choosing this material, however, consideration should be given to its impact on several areas other than weight. For example, cast aluminum will probably cost more than cast steel, so the tradeoff between increased initial cost and lower shipping weight should be investigated. The durability and potential lifetimes of aluminum and steel fireboxes should be estimated and compared; this should include flame-impingement erosion and long-term cycle fatiguing. Particular design attention must be given to effects of differential thermal expansion where dissimilar metals are joined. Prevention of excessive strain and yielding of one or the other metal, leading to development of gas leaks where an aluminum firebox is bonded to a steel heat exchanger, is an area of major concern. From a thermal or heat transfer standpoint, aluminum might be preferable to steel because of its higher conductivity; the combustor wall temperatures should be more uniform and slightly lower.

HEAT EXCHANGE CONSIDERATIONS

The photographs of the partially assembled prototype furnace in Fig. 6 show how the firebox and heat exchanger were installed and coupled. The firebox was mounted directly above the warm-air blower so that upward-flowing furnace coolant air flowed first over the finned hemispherical bottom of the firebox, then vertically upward between vertical fins on the firebox walls. Two sheet-metal baffles, one on either side of the firebox, prevented the warm air from bypassing the finned firebox. Altogether, the cross-sectional area for air flow past the combustor was reduced to about 58% of that in the stock Lennox furnace. Combined with the effect of air heating in this section of the furnace, the warm-air velocity past the top of the prototype firebox was increased by about 80% above that in the predecessor furnace. Even though the baffles flared out at that point, restoring the full nominal cross-sectional flow area through the heat exchanger, the air jets between baffles undoubtedly retained most of their elevated velocity and did not expand effectively to fill the available cross section. Thus, the outermost heat exchanger panels, above the flared ends of the baffles, were not cooled as effectively as were the inner panels. Additionally, the flanged joint between the firebox and the heat exchanger protruded about 3/4 inch into the air stream, tending to displace it away from the central cylindrical dome section of the heat exchanger and to reduce further the air-cooling effectiveness.

The design has been modified to help restore the effectiveness of the heat exchanger. The baffles in the warm-air passage have been moved further from the firebox, so that the flow cross section is less constricted. The baffles also begin to taper out at about the midsection of the firebox to permit the air flow to decelerate and expand to the heat exchanger cross section more gradually. The less bulky joint between the combustion chamber and heat exchanger also should contribute to smoother flow and effective heat exchange.

VESTIBULE CLOSURE PANEL

Only one modification to the exterior cabinet of the stock Lennox furnace was required. The optimum 1 ml/s (gph) residential oil burner has a longer blast tube than has the stock Lennox burner. As a result, the burner protruded beyond the front of the burner vestibule and interfered with the panel that closes the vestibule. For the integrated furnace design, a vestibule closure panel has been provided with a bulge to accommodate the burner.

SECTION VI

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7. Hall, R. E., J. H. Wasser, and E. E. Berkau: "A Study of Air Pollutant Emissions from Residential Heating Systems," EPA 650/2-74-003, Environmental Protection Agency, Research Triangle Park, North Carolina, January 1974.
8. "Standard for Safety: Oil-Fired Central Furnaces," U.L. Standard 727, Underwriters' Laboratories, Inc., Chicago, Illinois, November 1973.

APPENDIX A

FLUE GAS COMPOSITIONAL ANALYSIS

The sample flow train used for analyzing flue gas composition is illustrated in Fig. A-1. A 0.006 m (1/4 inch) diameter stainless-steel tubing sample probe was inserted near the combustor or flue pipe centerline, downstream of the heat exchanger. Flue gas aspirated into the sample probe flowed through a line to an air-cooled condensibles trap where particulates and heavy oils were separated out. Next, the gas passed into an ice-cooled, stainless-steel condensibles trap where most of the water and any condensible, low-volatility hydrocarbons were removed. After the condenser, the gas passed into a Pyrex wool-filled glass cylinder which served as a final separator for heavy oils and particulates, and provided a visual indication of the cleanliness of the gas being admitted to the analysis instruments. Table A-1 gives a summary of the gas analysis instruments used. The gas leaving the glass-wool filter was split into three parallel paths. One path led directly to the total hydrocarbon analyzer. A second path led through a Drierite bed where water vapor was removed, then into the series-plumbed CO, CO_2, and O_2 analyzers. The third path passed through a combined Drierite and 3 Å molecular sieve bed for total water removal, then into the nitric oxide analyzer. The gas was pumped through the system by three diaphragm pumps located downstream of the nitric oxide analyzer, total hydrocarbon analyzer, and the series of CO, CO_2 and O_2 analyzers.

When the analytical system shown in Fig. A-1 is used to analyze gases which may have been quenched before combustion was compled, there are two factors that must be considered in reducing the data: (1) only burned or partly pyrolyzed fuel is included in the analysis, since minute quantities of liquid or vapor fuel may be removed by the cold trap and (2) water formed from hydrogen and oxygen during the combustion process is also removed from the analyzed sample by the cold trap.

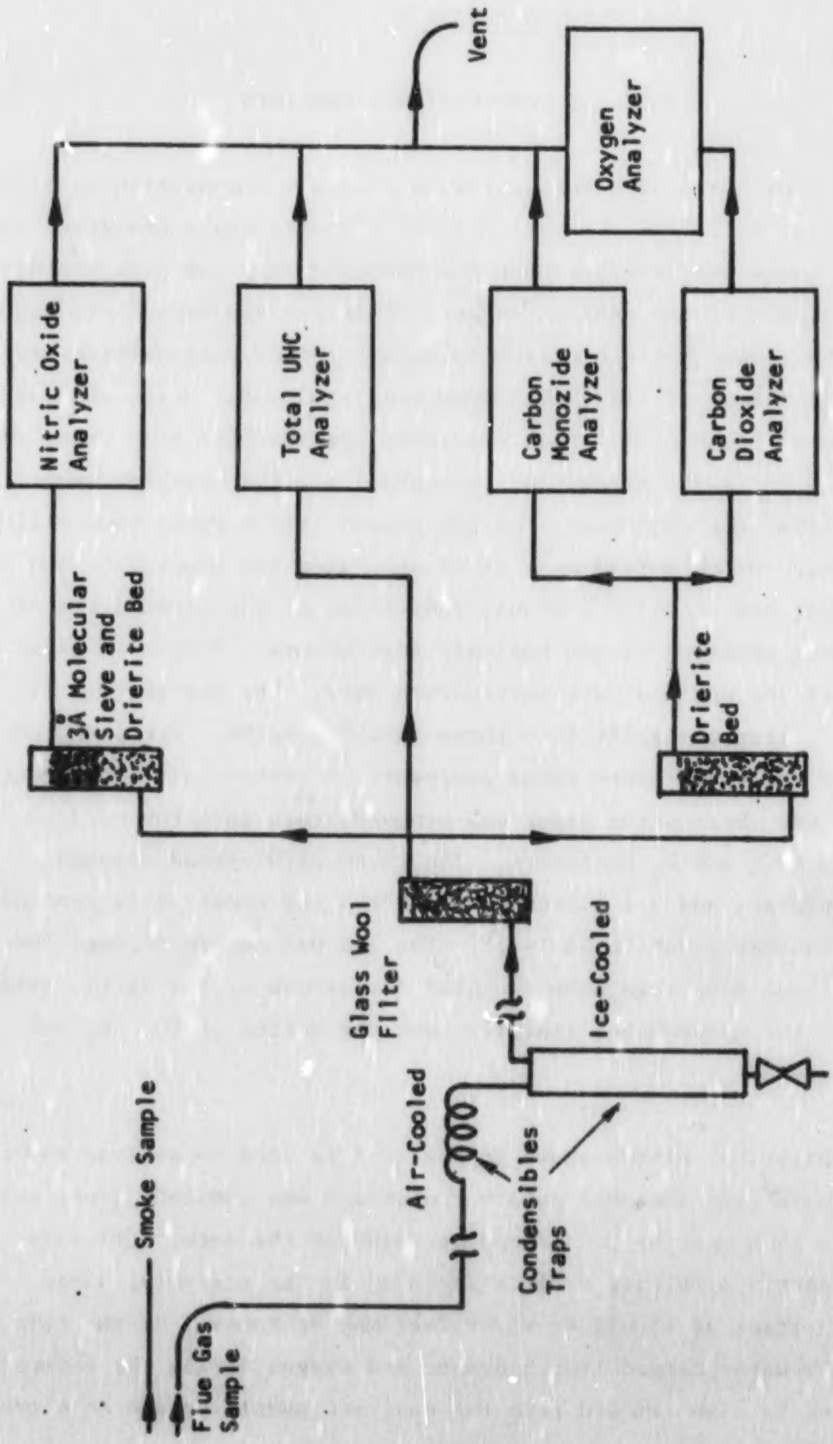
Flue Gas Sample
Smoke Sample
Air-Cooled
Condensibles Traps
Ice-Cooled
Glass Wool Filter
3Å Molecular Sieve and Drierite Bed
Drierite Bed
Nitric Oxide Analyzer
Total UHC Analyzer
Carbon Monozide Analyzer
Carbon Dioxide Analyzer
Oxygen Analyzer
Vent

Figure A-1. Analytical System for Fuel Oil Burner Emissions Analysis

Table A-1. EXHAUST ANALYSIS INSTRUMENTS

	CO	CO_2	NO	Total HC	Oxygen	Smoke
Type	MSA Nondispersive IR LIRA Model 300	MSA Nondispersive IR LIRA Model 300	MSA Nondispersive IR LIRA Model 200	MSA H_2 flame ionization detector	Beckman polarographic	Bacharach (manual)
Range	0 to 1500 ppm (mole)	0 to 20 mole %	0 to 500 ppm (mole)	0.2 to 800 ppm total HC by volume as CH_4	0 to 100%	0 to 9
Sensitivity	30 ppm minimum detectable	0.25% minimum detectable	10 ppm minimum detectable	10 ppm minimum detectable	~0.1%	1
Calibration	1000 ppm CO in N_2 standard gas	14% CO_2 in N_2 standard gas	0.82% C_2H_4 in N_2 used as simulant for 410-ppm NO standard	3% CH_4 in helium used as a standard	Air - 21% N_2 = 0%	Ten spots of monotonically varying darkness

Values calculated from the measured flue gas compositional data included the overall stoichiometric ratio, the weight of nitric oxide per unit weight of burned fuel, and the weight of carbon monoxide per unit weight of burned fuel. The method of calculation to obtain these values is described below.

The calculations were based on air having the following nominal composition:

Component	Mole %	Wt %
N_2	78.08	75.63
O_2	20.95	23.19
Noble gases (Ar, He and Ne)	0.94	1.13
CO_2	0.03	0.05
	100.00	100.00

The composition of the fuel was assumed to be characterized by the formula CH_x where, for the No. 2 fuel oil burned in this program, $x = 1.814$. The following symbols were used in the calculations:

AIR = moles of air to produce 100 moles of dry flue gas
FUEL = moles of fuel to produce 100 moles of dry flue gas
CO = moles of carbon monoxide in 100 moles of dry flue gas
CO_2 = moles of carbon dioxide in 100 moles of dry flue gas
NO = moles of nitric oxide in 100 moles of dry flue gas
O_2 = moles of oxygen in 100 moles of dry flue gas
HC = moles of hydrocarbon, as CH_4, in 100 moles of dry flue gas

The values of CO, CO_2, NO, O_2, and HC were obtained directly from the analysis instruments. In the following, it is assumed that all hydrogen is oxidized to water and condensed out of the system at the cold trap, prior to analysis.

An oxygen balance yields:

$$0.2095 \text{ AIR} = CO_2 - 0.0003 \text{ AIR} + 0.5 \text{ CO} + 0.25 \text{ x} (CO_2 + CO - 0.0003 \text{ AIR}) + 0.5 \text{ NO} + O_2 \qquad \text{(A-1)}$$

The left hand side of the above equation represents the total free oxygen contributed by the air. The first two items on the right side represent moles of oxygen tied up in CO_2, less the amount of CO_2 originally present in the air. The third term represents moles of oxygen tied up as carbon monoxide. The fourth term represents oxygen consumed to oxidize hydrogen, yielding the water condensed out in the cold trap. The fifth term is the oxygen tied up in nitric oxide. The sixth term is free oxygen remaining in the sample reaching the analysis instruments. Equation A-1 can be arranged to yield:

$$\text{AIR} = \frac{(1 + \frac{x}{4}) CO_2 + (1/2 + \frac{x}{4}) CO + 1/2 \text{ NO} + O_2}{0.2095 + 0.0003 + 0.0003 \text{ x}/4} \qquad \text{(A-2)}$$

A carbon balance can be used to calculate the moles of fuel burned per 100 moles of dry flue gas:

$$\text{FUEL} = CO_2 - 0.0003 \text{ AIR} + CO \qquad \text{(A-3)}$$

The moles of air available per mole of burned fuel in the sample gas can be obtained by taking the ratio of the values from Eq. A-2 and A-3. AIR must be calculated first, before calculation of FUEL. If the combustion were in stoichiometric proportions, the moles of air would be, by an oxygen demand calculation:

$$\text{AIR}_{\text{stoich}} = \frac{(1 + x/4) \text{ FUEL}}{0.2095} \qquad \text{(A-4)}$$

The stoichiometric ratio of the locally sampled burned gases is a parameter frequently used in this report. It is defined as the ratio of AIR to $\text{AIR}_{\text{stoich}}$ and is designated SR.

$$SR = \frac{AIR}{AIR_{stoich}} \tag{A-5}$$

Combination of Eq. A-2 through A-5 yields a direct calculation of the burned gas stoichiometric ratio in terms of the measured parameters:

$$SR = \frac{\dfrac{(1 + \frac{x}{4})\ CO_2 + (1/2 + \frac{x}{4})\ CO + 1/2\ NO + O_2}{0.2095 + 0.003 + 0.0003\ x/4}}{\dfrac{(1 + \frac{x}{4})}{0.2095}\left[CO_2 + CO - 0.0003\dfrac{(1 + \frac{x}{4})CO_2 + (1/2 + \frac{x}{4})\ CO + 1/2\ NO + O_2}{0.2095 + 0.0003 + 0.0003\ x/4}\right]} \tag{A-6}$$

According to the above definition, when the sample contains just a sufficient amount of air to oxidize all of the fuel in the sample to CO_2 plus condensed-out water, then SR = 1. As a second example, if there is twice the required amount of air for complete oxidation of the fuel, then SR = 2. Note that the stoichiometric ratio, as calculated from Eq. A-6 does not require that the products in the flue gas be in chemical equilibrium.

Note that the accuracy of the stoichiometric ratio calculation would be affected very little if all terms in Eq. A-6 containing the factors 0.0003 and NO were ignored. These factors represent the carbon dioxide originally present in free air, and the oxygen tied up in nitric oxide, respectively.

One partially questionable assumption made in the formulation of Eq. A-6 was that all hydrogen originally present in the fuel becomes oxidized to water and is removed in the cold trap. This was a necessary assumption, since there was no instrument available to measure the actual hydrogen content of the sample gas. The assumption is very good under the combined conditions of air-rich stoichiometric ratios (SR > 1) and chemical equilibrium. To test this assumption, a Rocketdyne thermochemical computer code was used to calculate the species concentrations under conditions of chemical equilibrium for stoichiometric ratios from 0.8 to 2.8. These calculations included the equilibrium presence of free H_2. The actual stoichiometric ratios of these combustion gases, compared to those calculated by Eq. A-6 (which does not recognize the presence of H_2) are given in Table A-2, where it can be seen that Eq. A-6 is quite accurate except for SR < 1. Calculated equilibrium conditions are tabulated in Tables A-3 and A-4.

Table A-2. VALIDITY OF STOICHIOMETRIC RATIO CONDITIONS

Actual Stoichiometric Ratio	Stoichiometric Ratio Calculated from Eq. B-6
0.800	0.844
1.000	1.003
1.200	1.197
1.400	1.400
1.600	1.600
2.000	2.002
2.400	2.404
2.800	2.804

The primary cause of the inaccuracy at SR < 1 is the unaccounted for presence of H_2. In nonequilibrium gases, there is likely to be H_2 present even where none would be indicated from equilibrium calculations and, at fuel-rich conditions, there could be more or less than indicated from equilibrium calculations. Because of this likelihood of nonequilibrium, no attempt was made to correct the calculations of Eq. A-6 by means of equilibrium calculations.

The concentration of CO_2 (dry basis) in the flue gas in the parameter most often used in the space heating industry as an indication of combustion conditions. To illustrate the relationship of $\%CO_2$ to the stoichiometric ratio, equilibrium data from Table A-4 were used to calculate the curve shown in Fig. A-2; a calculated $\%O_2$ curve is also shown. A number of values of measured CO_2 concentrations in actual

Table A-3. EQUILIBRIUM COMBUSTION GAS PROPERTIES FOR NO. 2 DISTILLATE FUEL OIL BURNED WITH AIR

($CH_{1.814}$, 18,443 Btu/lb Net Heat of Combustion With Air at 14.67 psia)

Stoith. Ratio*	Oil + Air Inlet Temp., F	Flame Temperature, F	C_p Frozen, Btu/lb-R	Y Frozen	Viscosity, centipoise	Thermal Conductivity, Btu/hr-ft-F	Prandtl Number	Molecular Weight
0.8	0	3429	0.346	1.261	0.0666	0.0702	0.7946	27.73
1.0		3614	0.341	1.254	0.0687	0.0711	0.7984	28.80
1.2		3290	0.333	1.260	0.0653	0.0661	0.7954	29.00
1.4		2940	0.324	1.267	0.0615	0.0610	0.7915	29.03
1.6 Air		2649	0.318	1.275	0.0581	0.0567	0.7880	29.03
2.0 Rich		2209	0.307	1.288	0.0527	0.0500	0.7820	29.02
2.4 ↓		1897	0.298	1.298	0.0487	0.0452	0.7771	29.01
2.8		1663	0.291	1.308	0.0456	0.0415	0.7730	29.00
0.8	70	3778	0.347	1.261	0.0671	0.0709	0.7948	27.72
1.0		3649	0.341	1.254	0.0691	0.0715	0.7984	28.77
1.2		3336	0.333	1.259	0.0658	0.0667	0.7956	29.00
1.4		2991	0.325	1.267	0.0621	0.0617	0.7918	29.03
1.6		2703	0.318	1.274	0.0589	0.0574	0.7884	29.03
2.0		2765	0.308	1.286	0.0535	0.0509	0.7825	29.02
2.4		1955	0.299	1.297	0.0495	0.0461	0.7778	29.01
2.8		1722	0.193	1.306	0.0464	0.0425	0.7738	29.00
0.8	200	3867	0.347	1.260	0.0681	0.0720	0.7951	27.71
1.0		3709	0.342	1.257	0.0698	0.0723	0.7983	28.73
1.2		3418	0.334	1.259	0.0668	0.0678	0.7958	28.98
1.4		3085	0.326	1.266	0.0632	0.0629	0.7923	29.02
1.6		2802	0.320	1.273	0.0600	0.0588	0.7890	29.02
2.0		2369	0.309	1.284	0.0548	0.0524	0.7834	29.02
2.4		2061	0.301	1.294	0.0509	0.0477	0.7790	29.01
2.8		1831	0.295	1.303	0.0479	0.0441	0.7751	29.00

*Stoichiometric ratio is unity at 14.49 masses of air per mass of fuel, and proportionately greater than unity for increasing relative mass of air.

Table A-4. CALCULATED EQUILIBRIUM COMBUSTION GAS COMPOSITION, VOLUME OR MOLE PERCENT

Stoich. Ratio	Oil + Air Inlet Temp., F	H	O	Ar	OH	H_2	H_2O	CO	CO_2	NO	N_2	O_2
0.8	0	0.0630	0.0000	0.821	0.0499	2.016	12.263	7.243	8.687	0.000	68.837	0.000
1.0		0.0397	0.0313	0.866	0.2816	0.250	11.690	1.393	12.052	0.253	72.522	0.619
1.2		0.000	0.0217	0.882	0.1862	0.030	10.141	0.161	11.247	0.390	73.784	3.160
1.4		0.000	0.0000	0.890	0.0757	0.000	8.832	0.0203	9.841	0.2955	74.465	5.566
1.6		0.000	0.0000	0.895	0.0790	0.000	7.799	0.000	8.679	0.2080	74.947	7.444
2.0		0.000	0.0000	0.902	0.000	0.000	6.297	0.000	7.000	0.0829	75.603	10.107
2.4		0.000	0.0000	0.907	0.000	0.000	5.276	0.000	5.864	0.0339	76.028	11.888
2.8		0.000	0.0000	0.910	0.000	0.000	4.541	0.000	5.046	0.000	76.326	13.161
0.8	70	0.0737	0.0000	0.821	0.0613	1.996	12.271	7.268	8.659	0.017	68.901	0.000
1.0		0.0455	0.0362	0.866	0.3072	0.269	11.647	1.501	11.934	0.272	72.456	0.666
1.2		0.0000	0.0261	0.882	0.2082	0.036	10.121	0.195	11.210	0.404	73.751	3.159
1.4		0.0000	0.0000	0.890	0.0885	0.000	8.824	0.026	9.835	0.322	74.447	5.553
1.6		0.0000	0.0000	0.895	0.0351	0.000	7.795	0.000	8.678	0.223	74.933	7.432
2.0		0.0000	0.0000	0.902	0.000	0.000	6.297	0.000	7.000	0.096	75.596	10.100
2.4		0.0000	0.0000	0.907	0.000	0.000	5.276	0.000	5.863	0.041	76.023	11.884
2.8		0.0000	0.0000	0.910	0.000	0.000	4.541	0.000	5.046	0.018	76.323	13.159
0.8	200	0.0964	0.0000	0.821	0.0878	1.964	12.273	7.318	8.604	0.027	68.796	0.000
1.0		0.0577	0.0468	0.864	0.3579	0.304	11.562	1.710	11.705	0.310	73.328	0.754
1.2		0.0000	0.0356	0.882	0.2533	0.048	10.078	0.270	11.127	0.451	73.683	3.162
1.4		0.0000	0.0000	0.890	0.1157	0.000	8.806	0.042	9.816	0.373	74.405	5.526
1.6		0.0000	0.0000	0.895	0.0493	0.000	7.787	0.000	8.672	0.268	74.905	7.406
2.0		0.0000	0.0000	0.902	0.0000	0.000	6.295	0.000	7.000	0.125	75.582	10.085
2.4		0.0000	0.0000	0.907	0.0000	0.000	5.276	0.000	5.863	0.059	76.015	11.876
2.8		0.0000	0.0000	0.910	0.0000	0.000	4.541	0.000	5.046	0.028	76.319	13.154

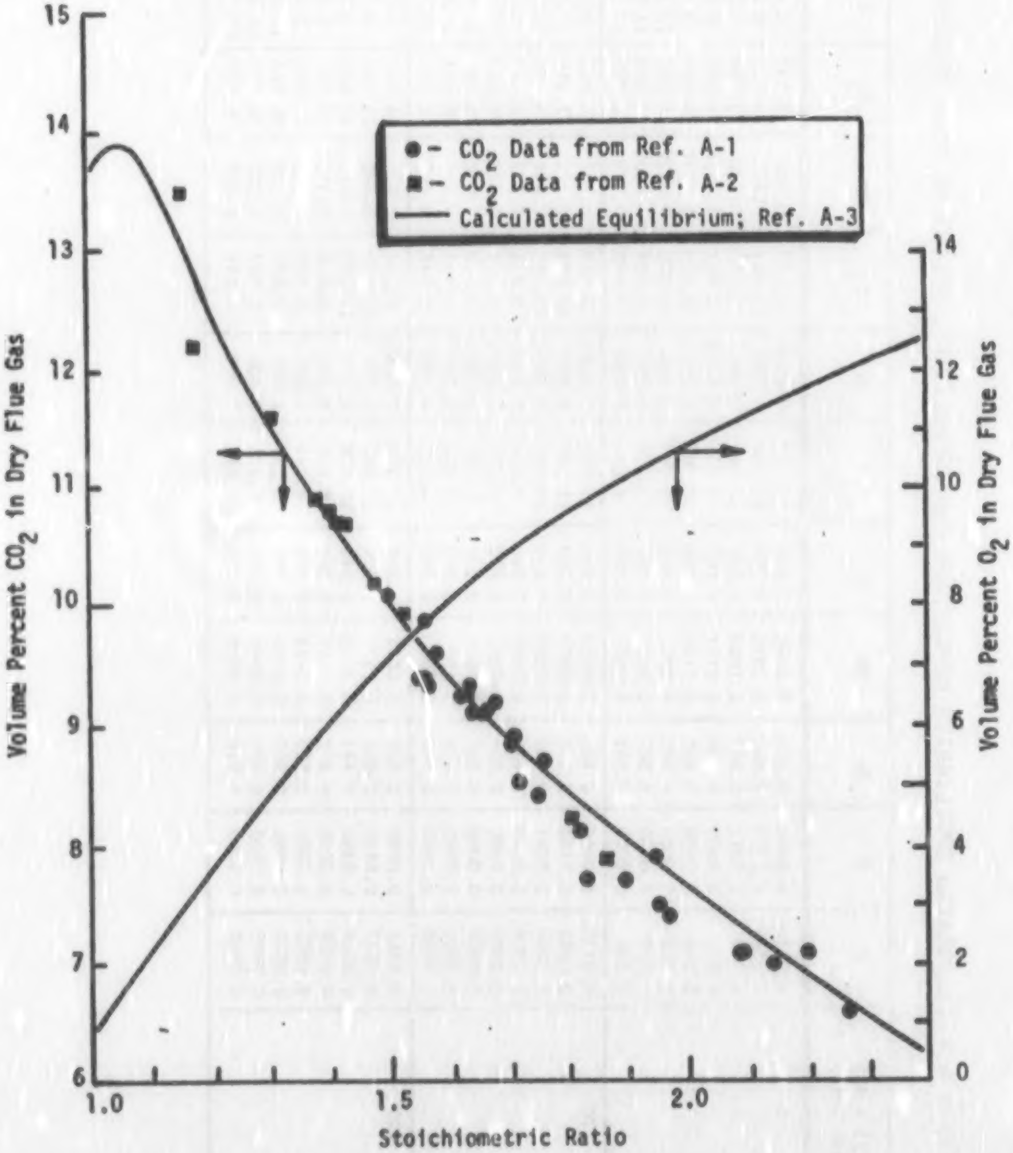

Figure A-2. Flue Gas CO_2 and O_2 Concentrations for No. 2 Fuel Oil Burned in Ambient Air at 1 atm furnace flue gases are also plotted on Fig. A-2. The measured data are seen to be very well correlated by the calculated equilibrium curve at SR > 1.1 (the calculated maximum CO_2 concentration as the stoichiometric condition is approached by reducing excess air is not normally observed in furnace testing).

Other parameters of interest for the flue gases are the mass ratio of nitric oxide to burned fuel, the mass ratio of carbon monoxide to burned fuel, and the mass ratio of unburned hydrocarbons (as CH_4) to burned fuel. These ratios are generally expressed herein as grams of nitric oxide per kilogram of burned fuel (g NO/kg fuel), grams of methane per kilogram of fuel (g UHC/kg fuel), and grams of carbon monoxide per kilogram of burned fuel (g CO/kg fuel). These parameters are calculated by aid of Eq. A-2 and A-3 from the following relationships:

$$\frac{\text{g NO}}{\text{kg fuel}} = \frac{(1000)\ (\text{NO})\ (MW_{NO})}{(CO_2 - 0.0003\ \text{AIR} + \text{CO})\ (MW_F)} \tag{A-7}$$

$$\frac{\text{g CO}}{\text{kg fuel}} = \frac{(1000)\ (\text{CO})\ MW_{CO}}{(CO_2 - 0.0003\ \text{AIR} + \text{CO})\ (MW_F)} \tag{A-8}$$

$$\frac{\text{g UHC}}{\text{kg fuel}} = \frac{(1000)\ (\text{HC})\ (MW_{CH_4})}{(CO_2 - 0.0003\ \text{AIR} + \text{CO})\ (MW_F)} \tag{A-9}$$

where

MW_{NO} = molecular weight of NO = 30.01

MW_F = molecular weight of fuel

= 12.01 + 1.008 x = 13.84

MW_{CO} = molecular weight of CO = 28.01

MW_{CH_4} = molecular weight of methane = 16.04

For calculation of the above quantities, the term 0.0003 AIR can be neglected without introducing more than about 0.1% error in the calculations, or AIR can be computed from Eq. A-3 and included in the calculation. The numbers given in this report include the effect of the term. The experimental data were reduced, according to the above equation, by means of a remote terminal timeshare computer program.

In addition to the gaseous pollutants described above, the smoke content of the mixed gases was also measured. The instrument utilized for this purpose was a Bacharach smoke meter. (It is manufactured by the Bacharach Instrument Company, Pittsburgh, Pennsylvania.) This is a hand-held device which, when pumped, sucks flue gases from a 0.006 m (1/4-inch) OD, uncooled sample probe through a piece of white filter paper; 10 strokes of the pump, over a period of about 15 seconds, causes the passage of 57.2 m^3 of flue gas per m^2 of filter paper (2250 $in.^3/in.^2$). The smoke particles deposit out on the filter paper. A reading is taken by comparing the darkness of the smoke deposition spot to a scale of 10 such calibrated spots provided with the instrument. The readings vary from 0 to 9. A reading of zero corresponds to no visually detectable deposit on the filter paper, while a reading of 9 corresponds to a dark black deposit. Intermediate readings are varying shades of black and gray, increasing in darkness with increasing reading numbers. A reading of 1 is generally accepted by the industry as a very acceptable degree of smoke. At the opposite extreme, a reading of 9, which is totally unacceptable, still does not correspond to sufficient smoke to be easily visible from observation of the exhaust stack outlet.

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APPENDIX B

DATA TABULATIONS: STOCK LENNOX FURNACE AND PROTOTYPE OPTIMUM FURNACE EXPERIMENTS

Experimental data are tabulated from laboratory tests of the stock Lennox 011-140 warm-air furnace prior to its conversion to the prototype optimum low-emission furnace and from tests of that latter unit in its initial and subsequently modified configurations. Two adjacent tables of data are given for each series of tests, one for flue gas concentration and air pollutant emissions data and the other for operational and thermal efficiency data. Some of the data are from steady-state experiments but most are from cyclical operation experiments. The tables are self-explanatory in this regard. Pollutant emissions data from cyclical testing were averaged over several (usually four) cycles while performance data were usually averaged over and recorded for each individual cycle.

Table B-1. CYCLE-AVERAGED POLLUTANT EMISSION DATA: STOCK LENNOX MODEL 011-140 FURNACE

(4-minute on/8-minute-off cycles)

STOCK LENNOX BURNER

	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
0.85-70°-A NOZZLE	40	1.40	10.4	7.4	17	125	5	0.36	2.672	0.057	1.2*	160
	41	1.37	11.1	6.0	30	152	4	0.55	2.980	0.042	3.0	147
	42	1.81	8.6	10.1	20	100	3	0.49	2.639	0.049	1.0*	185
	43	1.62	9.4	8.5	16	111	3	0.37	2.602	0.040	0.8	168
	44	1.62	9.4	8.5	15	110	3	0.33	2.577	0.040	0.9	168
	45	1.38	11.2	6.1	18	117	3	0.35	2.315	0.030	4.0	147
1.00-70°-A NOZZLE	46	1.51	10.3	7.6	15	106	2	0.30	2.305	0.021	1.0*	164
	47	1.62	9.7	8.7	12	115	2	0.26	2.692	0.020	0.9	185
	48	1.67	9.3	9.0	15	114	2	0.34	2.737	0.022	0.9	196
	49	1.37	11.3	6.0	20	106	2	0.36	2.079	0.019	2.0	155
	50	1.41	11.0	6.5	15	103	2	0.29	2.072	0.021	1.2*	162
	51	1.57	9.9	8.1	18	113	2	0.40	2.557	0.019	1.2*	183
	52	1.68	9.1	8.9	15	109	2	0.34	2.647	0.023	1.1*	194

SMOKE ONLY ON START, IMMEDIATE RECOVERY TO ZERO

OPTIMUM BURNER

	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
1.00-60°-A NOZZLE	74	1.33	11.7	5.5	13	97	2	0.25	1.832	0.020	0.0	185
	75	1.27	12.3	4.8	25	89	3	0.43	1.617	0.033	0.0	172
	76	1.32	11.9	5.5	17	97	2	0.32	1.832	0.020	0.0	186
	77	1.44	10.8	6.9	17	100	2	0.35	2.076	0.021	0.0	203
	78	1.67	9.4	9.1	11	110	4	0.27	2.650	0.050	0.0	226
	79	1.30	12.0	5.3	10	92	1	0.19	1.722	0.015	0.0	179
	80	1.47	10.6	7.2	15	103	1	0.30	2.165	0.015	0.0	207
	81	1.54	10.0	7.9	11	107	2	0.25	2.370	0.024	0.0	211
	82	1.57	9.9	8.2	11	105	1	0.25	2.382	0.016	0.0	215
	83	1.71	9.1	9.4	11	110	3	0.26	2.721	0.041	0.0	232
	84	1.35	11.4	6.3	15	95	2	0.28	1.877	0.021	0.0	197
	85	1.36	11.4	5.9	15	103	2	0.27	2.005	0.021	0.0	187
	86	1.69	9.1	9.1	11	112	3	0.21	2.737	0.035	0.0	229
	87	1.53	10.6	8.0	10	114	2	0.23	2.345	0.024	0.0	217

Table B-2. CYCLE-AVERAGED AND PSEUDO-STEADY-STATE EFFICIENCY DATA: STOCK LENNOX MODEL 011-140 FURNACE

(Listed by individual 4-minute-on/8-minute-off cycles)

STOCK LENNOX BURNER

0.85-70°-A NOZZLE

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
40	1.45	80.11	84.46	257	391	8392	6783	0.6114	23.9	148	30
40	1.45	81.15	84.36	239	403	7782	6315	0.5739	23.3	150	28
40	1.45	80.31	84.59	240	370	7819	6280	0.6165	23.4	146	30
40	1.45	78.68		243	383	7917	6224	0.6127	22.6		30
		80.05	84.47								
41	1.33	80.77	86.12	262	401	8320	6720	0.6127	23.3	135	30
41	1.33	76.79	86.30	257	365	8177	6279	0.6142	23.8	132	30
41	1.35	80.43	86.17	239	338	7576	6093	0.6207	24.7	133	26
41	1.38	77.30	86.06	240	332	7407	5881	0.6149	24.5	133	26
		78.82	86.16								
42	1.76	76.65	80.41	262	344	8615	6603	0.6447	25.3	188	23
42	1.76	76.76	80.41	258	352	8483	6511	0.6539	24.1	188	23
42	1.77	80.09	80.34	241	317	7924	6347	0.6702	25.4	188	23
42	1.76	78.74	80.63	241	331	7927	6241	0.6474	24.8	185	24
		78.06	80.45								
43	1.62	84.12	82.88	258	344	8275	6961	0.6434	26.7	163	24
43	1.59	81.97	82.86	239	334	7669	6287	0.6392	25.1	165	25
43	1.62	81.68	83.13	241	343	7736	6319	0.6484	24.2	158	26
43	1.59	77.70	83.05	245	341	7867	6112	0.6363	23.9	161	26
		81.37	82.98								
44	1.56	73.64	83.20	256	464	8763	6453	0.5038	23.5	160	30
44	1.60	73.75	83.07	239	434	8181	6033	0.5158	22.9	160	30
44	1.60	75.98	83.07	241	442	8250	6268	0.5191	23.3	160	30
44	1.64	79.34	83.16	246	494	8421	6681	0.4884	23.6	157	30
		75.69	83.13								
45	1.36	77.97	86.20	241	461	8053	6246	0.5182	22.2	132	30
45	1.35	90.60	86.42	244	463	8153	7390	0.5124	26.5	189	30
45	1.33	82.24	86.37	255	494	8524	7010	0.5113	23.6	131	31
45	1.33	74.98	86.12	258	455	8619	6462	0.5080	23.8	135	30
		81.36	86.28								

1.0-70°-A NOZZLE

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
46	1.42	80.53	83.87	241	287	9122	7345	0.8527	25.5	162	25
46	1.47	81.72	83.65	245	301	9273	7577	0.8635	24.8	159	25
46	1.46	80.89	83.91	263	323	9954	8051	0.8505	24.9	156	25
46	1.47	82.05	83.81	259	308	9696	7956	0.8545	25.7	156	25
		81.29	83.81								
47	1.57	70.75	81.53	241	242	9311	6588	0.8871	26.1	188	23
47	1.56	75.76	81.59	245	245	9466	7171	0.9506	26.1	188	23
47	1.53	72.19	81.87	263	300	10161	7335	0.8179	25.5	186	23
47	1.53	78.68		257	332	9929	7812	0.7895	25.3		23
		74.34	81.66								
48	1.62	75.19	80.50	242	271	9350	7030	0.8453	26.1	202	23
48	1.64										
48	1.62	72.05	80.73	264	317	10206	7354	0.8303	23.7	198	25
48	1.62	73.97	80.77	259	305	10016	7408	0.8304	24.9	198	25
48	1.62	72.06	80.60	241	278	9320	6715	0.8309	24.7	201	25
		73.32	80.65								
49	1.33	75.45	85.25	245	307	9267	6992	0.8319	23.3	149	24
49	1.33	76.67	85.07	263	307	9948	7627	0.8364	25.3	152	24
49	1.34	80.95	85.13	259	334	9790	7926	0.8193	24.7	150	23
49	1.31	76.92	85.30	240	296	9075	6981	0.8296	24.2	150	23
		77.50	85.19								
50	1.45	79.71	83.52	265	316	9873	7870	0.7372	28.8	165	17
50	1.38	74.80	84.04	259	302	9647	7216	0.6925	29.3	165	16
50	1.36	78.14		240	316	8953	6996	0.6622	28.5		19
50	1.38	85.01	84.11	240	311	8939	7616	0.7396	28.1	164	20
		79.42	83.89								
51	1.59	78.22	81.41	244	302	9307	7280	0.7330	27.9	189	20
51	1.51	75.69	81.78	263	323	10037	7597	0.7221	27.7	188	21
51	1.53	78.61	81.87	258	368	9846	7740	0.6460	27.7	186	21
51	1.51	76.36	81.99	239	311	9127	6969	0.7075	26.9	185	22
		77.22	81.76								
52	1.65	75.95	80.18	246	337	9489	7207	0.6934	26.3	205	21
52	1.65	74.66	80.55	262	338	10109	7548	0.7164	26.5	199	21
52	1.65	71.83	80.35	258	316	9952	7149	0.7187	26.8	202	21
52	1.65	72.70	80.52	240	300	9258	6730	0.7355	26.0	200	21
		73.79	80.40								

Table B-2 (concluded). CYCLE-AVERAGED AND PSEUDO-STEADY-STATE EFFICIENCY DATA: STOCK LENNOX MODEL 011-140 FURNACE

OPTIMUM BURNER

1.00–80°–A NOZZLE

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
74	1.26	86.33	82.97	263	347	9957	8597	0.8945	23.5	190	26
74	1.31	88.52	83.17	289	382	10942	9685	0.8933	24.2	183	26
74	1.29	80.56	82.96	282	375	10677	8601	0.8640	22.6	188	26
74	1.30	86.19	83.14	259	377	9815	8460	0.8996	21.2	185	27
		85.40	83.06								
75	1.16	89.66	84.77	263	403	9864	8844	0.8869	21.1	170	28
75	1.24	89.46	84.53	279	431	10465	9362	0.8919	20.7	167	28
75	1.24	83.25	84.56	274	434	10280	8556	0.8843	19.0	166	29
75	1.25	83.22	84.71	255	421	9567	7962	0.8786	18.3	163	29
		86.40	84.64								
76	1.29	83.82	83.36	256	373	9716	8144	0.8848	21.0	182	30
76	1.26	78.71	83.53	255	347	9681	7620	0.8619	21.7	181	31
76	1.29	79.76	83.33	235	341	8916	7112	0.8600	20.6	182	30
76	1.27	75.03	83.24	235	302	8916	6690	0.8725	21.6	155	30
		79.33	83.36								
77	1.41	76.43	81.19	240	308	10168	7771	0.8724	24.6	208	26
77	1.38	73.70	81.18	261	293	10207	7522	0.9027	24.2	211	26
77	1.40	75.00	81.21	241	268	9422	7066	0.9274	24.2	208	26
77	1.41	74.37	81.12	244	256	9536	7092	0.9464	25.0	209	25
		74.87	81.17								
78	1.60	66.37	78.03	235	236	9450	6272	0.9048	24.9	244	21
78	1.59	73.98	78.19	259	287	10422	7710	0.8815	25.9	243	22
78	1.59	71.74	77.94	260	274	10456	7501	0.8957	26.0	247	21
78	1.65	72.86	77.99	240	281	9658	7037	0.9225	23.1	240	22
		71.24	78.04								
79	1.25	75.21	83.53	260	284	9728	7316	0.8826	24.8	182	24
79	1.33	83.86	83.97	240	272	8974	7525	0.9116	23.8	170	23
79	1.27	78.93	83.87	238	263	8899	7024	0.8892	25.5	175	23
79	1.25	78.27	83.82	246	283	9201	7201	0.8828	24.5	177	23
		79.07	83.80								
80	1.42	79.97	81.40	260	380	10252	8199	0.9096	23.9	203	22
80	1.43	80.61	80.35	260	317	10256	8267	0.9434	23.5	217	23
80	1.42	86.94	80.19	241	226	9482	8243	1.2083	25.7	224	18
80	1.40	62.38	80.43	237	251	9324	5825	0.7651	25.8	221	18
		77.33	80.64								

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
81	1.51	69.77	79.42	258	304	10475	7308	0.8692	23.5	231	20
81	1.51	73.90	79.83	259	296	10508	7766	0.8779	25.4	224	18
81	1.50	72.01	79.69	239	281	9694	6980	0.8992	23.5	227	18
81	1.50	68.60	79.69	237	269	9613	6594	0.8892	23.4	227	18
		71.07	79.66								
82	1.53	68.56	79.39	260	265	10229	7013	0.8976	25.1	230	18
82	1.53	73.87	79.29	234	233	9197	6794	0.9490	26.1	231	17
82	1.51	67.01	79.17	234	217	9194	6161	0.9295	26.0	235	16
82	1.51	71.70	79.06	241	226	9476	6794	0.9653	26.5	236	17
		70.28	79.23								
83	1.65	71.86	77.15	256	251	10586	7607	0.9802	26.3	253	17
83	1.65	67.86	76.84	237	236	9804	6652	0.9416	25.4	258	17
83	1.65	70.75	77.19	234	254	9680	6848	0.9135	25.1	253	17
83	1.65	61.40	77.22	243	280	10055	6174	0.8484	22.1	252	18
		67.97	77.10								
84	1.38	73.91	81.67	256	283	10065	7439	0.8822	25.4	203	17
84	1.31	72.08	82.12	236	250	9279	6688	0.9100	25.0	200	17
84	1.29	69.93	82.56	233	268	9161	6406	0.9016	22.6	195	17
84	1.33	71.23	82.17	243	263	9551	6803	0.9135	24.1	198	17
		71.79	82.13								
85	1.33	79.71	82.83	240	259	9264	7384	1.0782	22.5	187	22
85	1.33	75.62	82.48	235	250	9071	6859	1.0920	21.4	193	22
85	1.31	79.27	82.61	243	274	9382	7437	1.0741	21.5	192	22
85	1.31	72.71	82.82	256	286	9887	7189	1.0476	20.4	189	23
		76.83	82.69								
86	1.65	71.31	77.71	241	272	9803	6991	1.0550	20.7	245	23
86	1.65	73.28	77.50	254	272	10329	7569	1.0870	21.7	248	22
86	1.65	74.13	77.75	253	293	10295	7631	1.0738	20.6	244	23
86	1.65	70.29		235	284	9565	6723	1.0551	19.1		24
		72.25	77.66								
87	1.50	72.77	80.18	240	283	9766	7107	1.0641	20.1	219	23
87	1.47	69.35	80.15	253	293	10298	7141	1.0696	19.4	221	24
87	1.39	71.88	78.47	253	301	10295	7399	1.0599	19.7	238	23
87	1.37	71.91	78.56	233	283	9481	6818	1.0587	19.4	238	23
		71.48	79.34								

Table B-3. STEADY-STATE POLLUTANT EMISSION DATA: PROTOTYPE OPTIMUM FURNACE WITH VARIOUS BURNERS

NOTE: 1.05 ml/s (1.00 gph) Firing Rate
0.57 m^3/s (1200 cfm) Warm-Air Flowrate (except as noted)

	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C	$\eta_{F.G.}$ %
OPTIMUM BURNER	149	1.20	13.0	3.7	30	38	5	0.48	0.653	0.047	2.1	266	80.76
	150	1.41	11.0	6.5	42	35	17	0.81	0.703	0.182	0.2	282	78.69
	151	1.48	10.4	7.3	57	45	21	1.15	0.968	0.238	0.0	291	77.64
	152	1.15	13.4	3.0	80	51	21	1.22	0.846	0.183	0.7	257	81.76
	153	1.22	12.7	4.0	25	46	4	0.42	0.804	0.037	0.5	266	80.88
	154	1.22	12.6	4.0	32	48	5	0.53	0.847	0.046	0.6	254	81.19
WILLIAMSON BURNER	155	1.10	14.1	2.1	20	93	1	0.29	1.456	0.005	0.5	263	81.95
	156	1.07	14.5	1.4	100	85	2	1.41	1.288	0.015	2.5	257	82.56
	157	1.20	13.1	3.7	15	97	1	0.24	1.646	0.008	0.0	272	80.75
	158	1.27	12.4	4.8	10	93	2	0.17	1.679	0.017	0.0	283	79.95
	159	1.32	11.9	5.5	10	88	2	0.19	1.662	0.018	0.0	290	79.40
	160	1.39	11.3	6.4	10	84	1	0.20	1.674	0.013	0.0	293	78.65
	161	1.21	12.9	4.0	11	96	2	0.19	1.659	0.016	0.0	277	80.60
	162	1.14	13.8	2.7	17	95	2	0.27	1.531	0.016	0.0	268	81.74
LENNOX BURNER	163	1.49	10.6	7.5	10	86	1	0.20	1.832	0.009	0.0	282	78.47
	164	1.45	10.8	7.0	10	90	0	0.19	1.864	0.003	0.0	279	78.92
	165	1.36	11.4	6.0	10	92	0	0.18	1.798	0.003	1.5	272	79.71
	166	1.29	12.0	5.1	15	92	0	0.26	1.699	0.003	4.0	264	80.40
	167	1.54	10.2	8.0	7	85	0	0.17	1.886	0.001	0.0	288	77.69
	168	1.51	10.3	7.7	7	86	0	0.16	1.866	0.001	0.0	288	77.81
	169	1.47	10.6	7.2	10	87	0	0.20	1.841	0.003	0.0	283	78.44
	170	1.43	10.9	6.8	10	90	0	0.19	1.842	0.003	1.5	278	79.10
OPTIMUM BURNER (0.460 m^3/s)	171	1.28	12.2	4.9	20	59	0	0.36	1.080	0.003	0.0	318	78.20
	172	1.24	12.6	4.3	20	57	0	0.33	1.012	0.002	0.0	313	78.71
	173	1.22	12.8	4.0	20	55	0	0.34	0.964	0.005	0.0	313	78.85
	174	1.35	11.5	5.9	18	59	0	0.34	1.143	0.005	0.0	321	77.42
	175	1.32	11.7	5.5	20	59	0	0.35	1.122	0.005	0.0	321	77.65
	176	1.26	12.4	4.6	20	56	0	0.33	1.013	0.005	0.0	317	78.44

Table B-4. STEADY-STATE EFFICIENCY DATA: PROTOTYPE OPTIMUM FURNACE WITH VARIOUS BURNERS

	RUN NO.	STOIC. RATIO	CO2 %	$^{\eta}$F.G. %		RUN NO.	STOIC. RATIO	CO2 %	$^{\eta}$F.G. %
---OPTIMUM BURNER---	149	1.20	13.0	80.76	---LENNOX BURNER---	163	1.49	10.6	78.47
	150	1.41	11.0	78.69		164	1.45	10.8	78.92
	151	1.48	10.4	77.64		165	1.36	11.4	79.71
	152	1.15	13.4	81.76		166	1.29	12.0	80.40
	153	1.22	12.7	80.88		167	1.54	10.2	77.69
	154	1.22	12.6	81.19		168	1.51	10.3	77.81
---WILLIAMSON BURNER---	155	1.10	14.1	81.95		169	1.47	10.6	78.44
	156	1.07	14.5	82.56		170	1.43	10.9	79.10
	157	1.20	13.1	80.75	---OPTIMUM BURNER--- (0.460 m^3/s)	171	1.28	12.2	78.20
	158	1.27	12.4	79.95		172	1.24	12.6	78.71
	159	1.32	11.9	79.40		173	1.22	12.8	78.85
	160	1.39	11.3	78.65		174	1.35	11.5	77.42
	161	1.21	12.9	80.60		175	1.32	11.7	77.65
	162	1.14	13.8	81.74		176	1.26	12.4	78.44

NOTES: 1.05 ml/s (1.00 gph) Firing Rates
0.57 m^3/s (1200 cfm) Warm-Air Flowrate (except as noted)

Table B-5. CYCLE-AVERAGED POLLUTANT EMISSION DATA: PROTOTYPE OPTIMUM FURNACE WITH SOME MINOR MODIFICATIONS

		RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
45° REAR BAFFLE	1.0-80°-A	177	1.42	11.1	6.8	10	54	0	0.19	1.118	0.001	0.0	288
		178	1.31	12.0	5.4	10	54	0	0.17	1.007	0.001	0.0	279
		179	1.27	12.4	4.8	10	52	0	0.17	0.948	0.001	0.0	274
		180	1.21	12.9	4.0	11	51	0	0.19	0.883	0.001	0.0	270
		181	1.16*	13.5	3.1	22	52	1	0.35	0.856	0.009	0.0	263
		182	1.47	11.1	7.5	11	54	0	0.24	1.154	0.001	0.0	288
		183	1.30	12.0	5.2	17	54	1	0.31	1.015	0.012	0.0	278
	1.0-80°-A	198	1.22	12.6	4.0	17	46	0	0.29	0.805	0.003	0.0	268
		199	1.16*	13.3	3.0	20	45	0	0.32	0.747	0.003	1.5	263
		200	1.18*	13.1	3.5	18	45	0	0.30	0.766	0.002	0.0	266
		201	1.27	12.1	4.7	11	46	0	0.20	0.840	0.001	0.0	274
		202	1.39	11.1	6.3	13	46	0	0.26	0.916	0.001	0.0	282
1.0-80°-A NOZZLE		203	1.17*	13.1	3.2	17	45	1	0.28	0.756	0.005	0.0	288
		204	1.12*	13.8	2.4	22	46	1	0.34	0.739	0.013	0.0	285
		205	1.09*	14.1	1.8	157	45	45	2.27	0.695	0.370	0.0	281
		206	1.29	12.0	5.0	13	45	1	0.24	0.835	0.010	0.0	298
		207	1.42	11.0	6.7	17	43	1	0.34	0.878	0.011	0.0	310
		208	1.23	12.6	4.2	15	44	1	0.26	0.780	0.011	0.0	299
		209	1.20	12.8	3.7	13	45	1	0.22	0.768	0.009	0.0	296
		210	1.13*	13.5	2.6	27	45	1	0.42	0.732	0.013	0.0	289

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* INTERMITTENT COMBUSTION RUMBLING

NOTE: Optimum Burner 1.05 ml/s (1.00 gph) Firing Rate

Table B-6. PSEUDO-STEADY-STATE EFFICIENCY DATA: PROTOTYPE OPTIMUM FURNACE WITH SOME MINOR MODIFICATIONS

		RUN NO.	STOIC. RATIO	CO2 %	$\eta_{F.G.}$ %
45° REAR BAFFLE	1.0-80°-A	177	1.42	11.1	78.49
		178	1.31	12.0	79.62
		179	1.27	12.4	80.10
		180	1.21	12.9	80.63
		181	1.16	13.5	81.43
		182	1.47	11.1	78.49
		183	1.30	12.0	79.68
	1.0-80°-A	198	1.22	12.6	
		199	1.16	13.3	
		200	1.18	13.1	
		201	1.27	12.1	
		202	1.39	11.1	

	RUN NO.	STOIC. RATIO	CO2 %	$\eta_{F.G.}$ %
1.0-80°-A	203	1.17	13.1	
	204	1.12	13.8	
	205	1.09	14.1	
	206	1.29	12.0	
	207	1.42	11.0	
	208	1.23	12.6	
	209	1.20	12.8	
	210	1.13	13.5	

NOTE; Optimum Burner, 1.05 ml/s (1.00 gph) Firing Rate

Table B-7. CYCLE-AVERAGED EMISSION DATA: PROTOTYPE OPTIMUM FURNACE WITH A REFINED DESIGN OPTIMUM LOW-EMISSION BURNER

		RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
	1.0-90°-A Nozzle	452	1.28	12.0	4.9	40	37	12	0.68	0.678	0.115	0.1	232
		453	1.25	12.3	4.4	41	38	7	0.70	0.682	0.066	0.0	227
		454	1.19	12.9	3.6	42	37	10	0.68	0.643	0.090	0.0	229
		455	1.16	13.2	3.0	50	37	14	0.76	0.622	0.122	0.0	227
		456	1.07	14.2	1.5	315	39	85	4.48	0.598	0.689	0.1	216
		457	1.29	11.9	5.0	40	38	12	0.70	0.713	0.117	0.0	224
0.0826 m Dia. Static Disc. + Extended Blower Lip	1.0-90°-A	458 - 461 Clogged Nozzle											
		462	1.30	11.7	5.0	30	50	10	0.52	0.927	0.098	1.7	243
		463	1.12	11.9	2.1	85	40	9	1.26	0.649	0.076	1.9	229
		464	1.16	13.1	3.1	45	43	6	0.69	0.713	0.053	1.5	236
		465	1.25	12.2	4.4	25	47	13	0.41	0.851	0.123	1.6	235
	1.0-60°-A	466	1.12	13.5	2.4	37	39	9	0.56	0.634	0.076	0.0	238
		467	1.15	13.3	2.8	30	39	9	0.45	0.647	0.078	0.0	241
		468	1.19	12.8	3.4	31	40	20	0.50	0.679	0.179	0.0	243
	1.0-70°-A Nozzle	469	1.30	11.8	5.1	37	38	25	0.66	0.710	0.246	0.0	243
		470	1.26	12.1	4.6	30	38	10	0.50	0.689	0.096	0.0	243
		471	1.22	12.5	3.9	22	38	7	0.37	0.671	0.064	0.0	243
		472	1.05	13.9	1.0	-1600	34	2500	-22.13	0.512	19.759	2.3	210
		473	1.18	12.9	3.4	23	40	7	0.38	0.684	0.067	0.3	227
		474	1.10	13.8	2.0	75	38	7	1.09	0.599	0.058	0.0	227
		475	1.14	13.3	2.7	45	38	7	0.68	0.621	0.065	0.0	227
		476	1.12	13.6	2.3	55	42	5	0.81	0.676	0.046	0.2	227
		477	1.20	12.6	3.7	31	38	7	0.51	0.655	0.064	0.0	235
	Early Warm-Air Cut-off	478	1.20	12.6	3.6	30	45	7	0.48	0.774	0.068	0.2	246
		479	1.12	13.7	2.3	50	45	12	0.74	0.721	0.101	1.4	235
		430	1.26	12.2	4.6	31	46	11	0.53	0.634	0.105	0.0	246
		481	1.29	11.5	4.9	41	32	11	0.72	0.599	0.108	0.0	257

NOTE: 4-minute-on/8-minute-off cycles
1.05 ml/s (1.00 gph) firing rate

Table B-7 (concluded). CYCLE-AVERAGED EMISSION DATA: PROTOTYPE OPTIMUM FURNACE WITH A REFINED DESIGN OPTIMUM LOW-EMISSION BURNER

1.0-70°-A Oil Nozzle, 0.0826m Dia. Static Disc + Extended Blower Lip

	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGF	NO GM/KGF	UHC GM/KGF	BACH. SMOKE	TFG C
Pyroflex Top & Bottom	482	1.26	12.4	4.6	21	61	4	0.37	1.102	0.034	2.6	252
	483	1.09	14.3	1.9	426	53	90	6.15	0.777	0.741	3.3	241
	484	1.19	13.1	3.6	42	54	3	0.68	0.927	0.027	3.0	249
	485	1.32	11.7	5.5	23	57	5	0.42	1.079	0.050	1.7	266
Pyroflex Top Only	486	1.07	14.5	1.4	535	36	185	7.55	0.558	1.006	1.5	235
	487	1.26	12.3	4.6	17	44	10	0.30	0.790	0.095	1.0	257
	488	1.19	12.8	3.6	20	39	5	0.32	0.675	0.045	0.0	238
	489	1.13	13.3	2.5	37	40	4	0.57	0.646	0.034	0.0	241
	490	1.38	11.1	6.1	45	48	17	0.83	0.952	0.183	1.2	257
Rear Baffle Installed	512	1.19	12.7	3.5	100	43	5	1.57	0.729	0.045	0.0	218
	513	1.25	13.8	5.0	80	43	5	0.50	0.768	0.047	0.0	238
	514	1.37	11.3	6.0	32	43	10	0.60	0.842	0.104	0.0	241
	515	1.24	12.2	4.2	45	43	4	0.74	0.760	0.042	0.0	232
Steady State	516	1.15	13.3	3.0	91	43	4	1.41	0.708	0.035	0.0	254
	517	1.06	14.3	1.4	1131	40	185	15.93	0.609	1.006	1.5	250
	518	1.40	11.1	6.4	20	46	0	0.37	0.918	0.005	0.0	271
	519	1.28	12.1	4.9	30	45	0	0.51	0.829	0.005	0.0	266
	520	1.22	12.7	4.0	50	44	0	0.81	0.771	0.005	0.0	261
	521	1.22	12.6	4.0	27	39	7	0.45	0.689	0.062	0.0	243
	522	1.09	14.1	1.8	620	35	60	8.92	0.539	0.656	0.5	235
	523	1.33	11.8	5.6	31	39	5	0.56	0.745	0.055	0.0	252
	524	1.14	13.5	2.7	65	38	4	0.98	0.619	0.039	0.0	238
High Vane Air Flow, 0.707 m³/s	525	1.25	12.4	4.4	20	35	4	0.35	0.619	0.038	0.0	228
	526	1.16	13.2	3.0	35	36	5	0.54	0.606	0.042	0.0	229
	527	1.18	13.5	2.4	80	38	7	1.19	0.604	0.059	0.0	229
	528	1.36	11.2	5.8	30	35	9	0.54	0.896	0.093	0.0	244
[illegible] Test	529	1.17	12.7	3.1	20	38	4	0.31	0.643	0.035	0.0	254
	529	1.17	13.1	3.3	29	38	5	0.54	0.639	0.041	0.1	242
[illegible] Chamber	530	1.37	11.3	6.1	45	36	8	0.27	0.712	0.083	0.0	257
	531	1.38	11.5	5.3	30	35	8	0.38	0.667	0.080	0.0	257
	532	1.28	11.9	4.6	27	36	6	0.45	0.649	0.038	0.0	257
	533	1.05	14.3	2.1	1099	33	190	15.87	0.493	0.953	1.1	

Table B-8. CYCLE-AVERAGED AND PSEUDO-STEADY-STATE EFFICIENCY DATA: PROTOTYPE OPTIMUM FURNACE WITH A REFINED DESIGN OPTIMUM LOW-EMISSION BURNER

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(M) F.G. C	T AMB C
452	1.25	80.84	81.66	224	331	8267	6633	0.5916	28.8	212	20
452	1.25	75.91	81.66	218	323	8046	6108	0.5616	28.6	212	20
452	1.25	81.58	81.84	229	346	8452	6890	0.5798	29.2	209	20
452	1.27	79.21	81.56	241	343	8895	7046	0.5748	30.4	212	20
		79.88	81.68								
453	1.22	91.95	82.30	220	446	8387	8156	0.5904	26.3	205	21
453	1.22	87.06	82.30	244	418	8911	7736	0.6008	26.3	205	21
453	1.22	81.83	82.30	244	367	8911	7292	0.6072	27.9	205	21
453	1.22	87.95	82.30	223	346	8144	7163	0.6436	27.4	205	21
		88.70	82.30								
454	1.16	83.23	82.95	223	371	8285	6898	0.5848	27.1	203	20
454	1.16	77.99	82.97	220	343	8213	6487	0.5887	27.4	203	20
454	1.16	73.99	82.64	230	349	8586	6353	0.5913	26.2	208	20
454	1.18	81.31	82.49	243	373	9071	7376	0.5862	28.7	208	20
		79.38	82.76								
455	1.12	80.94	83.16	219	356	8804	6640	0.5903	26.9	205	21
455	1.13	81.40	83.10	227	376	8477	6908	0.5807	26.9	204	21
455	1.13	77.94	83.04	243	403	9074	7072	0.5801	25.8	205	21
455	1.13	74.98	82.94	243	400	9074	6795	0.5541	26.1	208	21
		78.81	83.06								
456	1.03	82.37	84.06	223	389	8745	7309	0.6005	31.4	202	13
456	1.04	83.17	83.87	240	389	9412	7828	0.6061	33.4	205	13
456	1.04	84.38	84.11	222	334	8706	7347	0.6003	31.2	199	13
456	1.01	80.91	84.38	220	300	8489	6981	0.6033	32.9	199	13
		83.01	84.11								
457	1.25	80.87	81.63	244	358	8988	7815	0.6030	28.5	212	16
457	1.26	83.01	81.92	241	388	8876	7369	0.5876	29.8	207	16
457	1.25	87.71	81.98	223	341	8215	7205	0.6035	29.8	207	16
457	1.27	88.90	82.07	216	341	7957	7073	0.6062	29.1	204	16
		84.97	81.90								
462	1.29	75.06	80.43	223	269	9272	6959	0.6807	35.4	232	11
		75.06	80.43								
463	1.08	80.75	82.87	211	274	8495	7021	0.6315	34.5	218	11
		80.75	82.87								
464	1.14	71.51	81.96	220	265	9475	6776	0.6848	34.8	225	10
		71.51	81.96								
465	1.23	74.88	81.09	243	296	10010	7435	0.6844	34.2	224	10
		74.88	81.09								
466	1.10	81.37	82.30	220	305	9380	7747	0.6106	33.2	223	12
		81.37	82.30								
467	1.13	83.72	81.91	218	314	9408	7872	0.6804	34.4	229	11
		83.72	81.91								
468	1.17	82.12	81.44	223	309	9650	8021	0.6182	33.9	231	12
		82.12	81.44								
469	1.22	78.34	80.46	230	274	9097	7126	0.6146	36.0	233	10
		78.34	80.46								
470	1.24	78.36	80.73	241	290	9535	7471	0.6258	35.0	232	10
		78.36	80.73								
471	1.20	84.63	81.05	242	313	9574	8103	0.6164	35.7	232	10
		84.63	81.05								
472	1.07	79.47	83.96	223	280	8828	7015	0.5209	34.3	199	11
		79.47	83.96								
473	1.16	83.34	82.40	218	292	8638	7199	0.6379	32.9	213	13
		83.34	82.40								
474	1.07	82.59	83.16	229	286	9032	7459	0.6081	35.2	213	12
		82.59	83.16								
475	1.13	77.91	82.69	226	284	8953	6974	0.6169	33.8	213	12
		77.91	82.69								
476	1.09	87.89	83.30	214	317	8669	7618	0.6139	33.3	207	16
476	1.09	86.73	83.11	220	320	8920	7736	0.6277	32.7	211	17
476	1.09	86.87	83.17	229	317	9276	8058	0.6350	34.0	210	16
476	1.08	79.21	83.08	240	332	9719	7689	0.6037	32.6	213	15
		85.17	83.17								
477	1.19	73.43	81.84	224	338	8982	6596	0.5871	28.3	218	16
477	1.19	76.60	81.72	219	304	8779	6724	0.6054	31.1	219	15
477	1.19	88.96	81.90	216	300	8653	7697	0.6472	33.8	217	14
477	1.20	88.57	81.70	242	314	9707	8597	0.6667	34.9	218	16
		81.89	81.79								
478	1.19	73.70	81.21	219	199	8664	6385	0.6471	42.2	232	10
478	1.18	70.81	81.29	228	230	9020	6387	0.5639	41.8	232	10
478	1.19	75.13	81.18	243	227	9686	7232	0.6792	39.8	233	12
478	1.19	76.93	81.18	242	232	9586	7375	0.6667	40.6	233	12
		74.14	81.22								
479	1.08	74.91	82.60	218	209	8716	6530	0.6459	39.8	223	11
479	1.08	76.74	82.46	229	227	9162	7031	0.6718	39.2	226	12
479	1.07	70.11	82.60	243	233	9716	6812	0.6826	39.9	225	11
479	1.06	71.75	82.76	241	241	9636	6913	0.6148	39.7	225	11
		73.38	82.61								
480	1.24	75.29	80.94	216	233	8661	6521	0.6197	38.3	227	16
480	1.23	73.82	80.92	231	241	9263	6838	0.6295	38.3	230	16
480	1.21	76.57	80.99	243	268	9741	7458	0.6086	38.9	232	15
480	1.24	70.99	80.77	240	256	9624	6832	0.5960	38.1	231	16
		74.17	80.91								
481	1.31	79.77	79.99	226	254	8984	7167	0.6257	38.3	241	18
481	1.30	77.63	80.16	241	272	9581	7437	0.6146	37.8	238	18
481	1.32	81.72	80.04	244	300	9700	7927	0.5938	37.9	238	18
481	1.32	77.97	80.16	225	260	8944	6974	0.6052	37.6	235	18
		79.27	80.09								

NOTES: Burner fired at 1.05 ml/s (1.00 gph) in 4-minute-on/8-minute-off cycles

Table B-8 (concluded). CYCLE-AVERAGED AND PSEUDO-STEADY-STATE EFFICIENCY DATA: PROTOTYPE OPTIMUM FURNACE WITH A REFINED DESIGN OPTIMUM LOW-EMISSION BURNER

RUN NO.	STOIC RATIO	GROSS EFF. W.A. %	GROSS EFF. F.G. %	BURN TIME SEC	W.A. TIME SEC	Q FUEL KJ	Q W.A. KJ	WARM AIR M3/S	W.A. DEL-T C	T(N) F.G. C	T AMB C
482	1.21	72.65	80.82	243	254	9738	7074	0.6004	39.4	236	15
		72.65	80.82								
483	1.08	74.69	83.00	224	209	8968	6698	0.6636	41.0	227	13
		74.69	83.00								
484	1.14	84.33	81.58	227	214	9094	7668	0.6911	44.1	234	14
		84.33	81.58								
485	1.29	73.17	79.70	243	239	9731	7121	0.6227	40.6	251	13
		73.17	79.70								
486	1.01	67.57	83.48	219	202	8672	5860	0.6344	38.9	222	12
		67.57	83.48								
487	1.22	69.46	80.44	244	242	9865	6852	0.6080	38.3	244	12
		69.46	80.44								
488	1.17	74.69	81.73	241	269	9747	7279	0.6159	37.3	224	13
		74.69	81.73								
489	1.18	74.92	82.08	224	229	9056	6785	0.6891	40.1	227	12
		74.92	82.08								
490	1.35	65.61	79.38	230	217	9296	6099	0.6131	39.0	245	12
		65.61	79.38								
512	1.18	84.31	83.02	218	254	8481	7156	0.6162	38.8	199	18
		84.31	83.02								
513	1.25	78.31	81.17	229	236	8910	6978	0.6345	39.6	220	17
		78.31	81.17								
514	1.33	84.03	80.70	244	260	9500	7982	0.6153	39.4	221	18
		84.03	80.70								
515	1.23	83.28	81.86	222	250	8652	7205	0.6399	38.3	211	20
		83.28	81.86								
521	1.19	74.68	81.49	245	230	9429	7041	0.6132	39.1	226	17
521	1.19	85.58	81.49	219	244	8428	7213	0.6348	39.6	226	17
521	1.19	76.42	81.60	219	235	8428	6441	0.5929	39.3	223	17
521	1.19	84.93	81.49	225	263	8659	7354	0.5986	40.2	226	17
		80.40	81.52								
522	1.02	80.05	83.66	246	298	9492	7598	0.5971	36.3	213	21
522	1.08	80.97	83.52	242	290	9335	7558	0.6044	36.6	216	21
522	1.07	80.95	83.24	223	275	8602	6963	0.5884	36.6	213	21
522	1.06	90.62	83.30	216	283	8329	7548	0.6313	36.0	214	20
		83.15	83.43								
523	1.28	79.92	80.57	242	278	9434	7540	0.6181	37.3	230	21
523	1.28	80.32	80.59	240	295	9359	7517	0.5946	36.5	230	21
523	1.33	77.83	80.34	223	269	8696	6768	0.5873	36.4	230	21
523	1.28	79.82	80.48	219	265	8540	6817	0.6084	36.3	232	21
		79.47	80.49								
524	1.10	82.42	82.73	240	298	9365	7719	0.5968	36.9	217	22
524	1.12	83.71	82.54	241	310	9401	7870	0.6081	35.5	218	22
524	1.09	78.42	82.87	225	269	8777	6882	0.6151	35.3	216	22
524	1.09	69.82	82.98	219	263	8543	5963	0.6090	31.6	214	22
		78.59	82.78								
525	1.21	75.43	81.45	227	193	8762	6609	0.8382	34.7	221	22
525	1.21	94.11	81.65	240	232	9266	8721	0.8824	36.2	217	22
525	1.21	85.08	81.88	242	232	9350	7955	0.8894	32.8	213	23
525	1.21	82.98	81.98	224	209	8657	7183	0.8770	33.3	212	24
		84.40	81.74								
526	1.13	84.43	83.19	227	254	8784	7416	0.8343	29.7	202	26
526	1.13	81.23	83.34	241	250	9326	7575	0.8392	30.7	200	26
526	1.13	83.82	83.19	240	269	9287	7784	0.8260	29.8	202	26
526	1.13	85.42	83.32	223	257	8637	7378	0.8375	29.1	201	28
		83.72	83.26								
527	1.10	86.27	83.48	224	272	8676	7485	0.8246	28.3	203	28
527	1.10	85.18	83.72	243	302	9429	8031	0.8351	27.1	197	31
527	1.10	88.99	83.89	243	364	9432	8393	0.7719	25.4	194	32
527	1.10	81.92	84.08	219	355	8512	6973	0.7600	22.0	191	35
		85.59	83.76								
528	1.38	70.19	80.87	242	217	9268	6505	0.8754	29.1	216	27
528	1.35	79.21	80.78	244	212	9344	7401	0.9624	30.8	218	27
528	1.33	77.57	81.05	223	185	8540	6624	1.0305	29.5	216	27
528	1.33	76.40	80.87	218	172	8348	6378	1.0674	29.5	218	27
		75.84	80.89								
529	1.14	78.60	82.01	220	275	8919	7010	0.6117	35.4	223	19
529	1.14	80.17	81.94	241	284	9389	7527	0.6217	36.2	226	20
529	1.14	80.91	82.08	242	313	9434	7633	0.6097	34.1	222	21
529	1.13	76.73	82.04	225	256	8771	6730	0.5913	33.9	225	21
		79.10	82.02								
530	1.33	76.16	80.01	222	270	8880	6763	0.6173	33.5	237	19
		76.16	80.01								
531	1.31	75.68	80.12	240	281	9344	7072	0.6252	34.2	238	18
		75.68	80.12								
532	1.27	75.91	80.37	243	272	9467	7186	0.6371	35.2	237	20
		75.91	80.37								
533	1.02	79.41		223	257	8682	6894	0.6561	31.1		18
		79.41									

APPENDIX C

DATA TABULATIONS: EXPERIMENTAL FURNACE TESTS WITH HEAT EXCHANGER MODIFICATIONS

Experimental data are tabulated from several series of tests in which the prototype optimum and Williamson furnaces' heat exchangers were modified substantially. Emphasis in these experiments was on pollutant emissions reduction so only emissions-related data were measured and recorded. Both steady-state and cyclical tests were performed as denoted in the table titles.

One series of tests (Table C-3) involved forced recirculation of the flue gases back into the combustion chamber, so there is a table column labeled "Recirc. Ratio, %." That parameter was calculated from the formula given below the table title where SR_{mix} was estimated from calculated O_2, CO_2, and CO concentrations in a hypothetical mixture of the recirculated gases and the fresh reactants supplied to the burner. The recirculation ratio is defined as the relative mass of recirculated burned gases (at stoichiometric conditions) to the mass of unburned air (including unburned air in the recirculated gases); it indicates the relative dilution of combustion air with inert gaseous diluents.

Table C-1. CYCLE-AVERAGED POLLUTANT EMISSION DATA: FINNED-COMBUSTOR WILLIAMSON FURNACE CONFIGURATION

	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
1.0-60°-A	184	1.31	11.9	5.4	20	67	1	0.35	1.263	0.006	0.0	263
	185	1.26	12.4	4.6	20	68	0	0.33	1.220	0.005	0.0	257
	186	1.21	12.7	3.9	20	68	0	0.32	1.182	0.004	0.0	254
	187	1.13	13.5	2.6	22	69	0	0.34	1.111	0.003	1.5	248
	188	1.46	10.6	7.0	20	63	0	0.39	1.331	0.001	0.0	271
	189	1.42	10.8	6.6	20	63	0	0.38	1.299	0.002	0.0	270
	190	1.23	12.6	4.1	20	67	0	0.32	1.174	0.003	0.0	258
LO-80°-A	211	1.08	14.1	1.7	40	50	0	0.53	0.772	0.001	4.0	238
	212	1.15	13.4	3.0	20	50	0	0.31	0.823	0.001	2.5	249
	213	1.23	12.5	4.2	20	49	0	0.33	0.864	0.001	2.0	257
	214	1.42	10.9	6.6	20	45	0	0.38	0.924	0.001	1.0	268
	215	1.36	11.3	5.9	20	45	0	0.36	0.884	0.001	2.0	274
	216	1.14	13.4	2.7	25	52	0	0.38	0.849	0.001	3.0	257

NOTES: Optimum Burner Fired at 1.06 ml/s (1.00 gph) in 4 min. on/8 min. off cycles.

Table C-2. STEADY-STATE POLLUTANT EMISSIONS DATA: COIL-COOLED PROTOTYPE FURNACE WITH VARIOUS COMBUSTOR LENGTHS AND COOLING FLUIDS

	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
L=0.[illegible]m Water-Cooled	221	1.11	13.7	2.1	25	32	0	0.37	0.502	0.001	0.0	204
	222	1.09	13.9	1.8	30	31	0	0.43	0.487	0.003	0.0	202
	223	1.07	14.1	1.5	47	31	1	0.68	0.473	0.008	0.0	202
	224	1.05	14.4	1.0	110	30	6	1.52	0.454	0.048	0.0	199
	225	1.23	12.4	4.2	20	31	0	0.33	0.545	0.001	0.0	213
	226	1.20	12.6	3.7	20	30	0	0.32	0.526	0.002	0.0	210
	227	1.13	13.5	2.4	21	30	0	0.33	0.491	0.003	0.0	207
	228	1.15	13.3	2.9	20	31	0	0.31	0.508	0.003	0.0	203
Coil Removed	229	1.10	13.8	2.0	10	46	0	0.16	0.726	0.002	0.0	621
	229	1.12	13.8	2.4	10	46	0	0.15	0.739	0.002	0.0	621
	229	1.13	13.4	2.6	11	45	0	0.18	0.733	0.002	0.0	621
	230	1.03	14.8	0.6	80	45	2	1.09	0.658	0.016	2.0	621
	231	1.06	14.4	1.3	20	45	0	0.28	0.687	0.002	0.0	621
	232	1.16	13.3	3.0	11	45	0	0.18	0.747	0.002	0.0	621
	233	1.24	13.9	4.8	12	46	0	0.21	0.819	0.001	0.0	621
	234	1.12	13.8	2.3	11	45	0	0.18	0.721	0.002	0.0	621
L=0.5 m Air Cooled	235	1.25	12.2	4.5	60	45	0	1.00	0.813	-.000	0.0	338
	236	1.16	13.3	3.0	30	43	0	0.46	0.708	-.000	0.0	377
	237	1.08	13.9	1.6	30	44	0	0.43	0.683	-.000	0.0	377
	238	1.03	14.8	0.7	125	43	0	1.71	0.635	-.000	0.0	382
	239	1.02	14.9	0.4	-1600	41	200	-21.56	0.604	1.540	3.0	377
	240	1.13	13.5	2.5	20	46	0	0.30	0.736	-.000	0.0	404
	241	1.19	12.7	3.5	18	45	0	0.30	0.771	-.000	0.0	399

L=0.80m		RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
	Steam-Cooled	242	1.05	14.5	1.0	177	44	6	2.46	0.663	0.048	3.0	254
		243	1.08	14.1	1.7	40	43	1	0.57	0.671	0.005	2.5	257
		244	1.10	13.9	2.1	30	45	0	0.44	0.706	0.003	1.5	260
		245	1.13	13.6	2.5	20	45	0	0.30	0.721	0.002	1.5	264
		246	1.19	12.9	3.5	18	43	0	0.30	0.736	0.001	1.5	260
		247	1.29	12.0	5.0	15	48	0	0.27	0.886	0.001	0.0	257
		248	1.23	12.4	4.2	11	49	0	0.20	0.865	0.001	1.0	254
		249	1.07	14.1	1.7	40	41	1	0.59	0.634	0.007	2.5	243
	Water-Cooled	250	1.09	13.9	1.9	50	31	1	0.72	0.482	0.008	0.0	241
		251	1.35	11.4	5.8	20	31	0	0.38	0.598	0.001	0.0	238
		252	1.19	12.9	3.5	20	31	0	0.31	0.523	0.001	0.0	243
	Water-Cooled + Radiation Shield	253	1.31	11.5	5.1	20	37	1	0.35	0.705	0.006	0.0	288
		254	1.26	12.0	4.6	20	36	1	0.34	0.664	0.007	0.0	302
		255	1.20	12.5	3.7	20	35	1	0.32	0.615	0.008	0.0	310
		256	1.14	13.3	2.6	30	33	1	0.45	0.540	0.019	0.0	306
		257	1.07	13.9	1.5	110	31	3	1.56	0.473	0.028	0.0	299
		258	1.15	14.6	3.2	20	35	1	0.32	0.571	0.009	0.0	311
	Air-Cooled + Radiation Shield	259	1.16	13.1	3.0	21	47	0	0.34	0.787	0.001	0.0	376
		260	1.10	13.8	1.9	50	50	1	0.72	0.780	0.005	1.5	388
		261	1.06	14.1	1.2	139	49	4	1.95	0.741	0.029	3.0	391
		263	1.32	11.5	5.4	17	50	0	0.32	0.948	0.002	0.0	410
		264	1.27	12.0	4.7	20	50	0	0.34	0.908	0.002	0.0	416
		268	1.13	13.3	2.5	11	50	0	0.19	0.807	0.004	0.0	416

NOTE: Optimum burner fired at 1.05 ml/s (1.00 gph)

Table C-2 (concluded). STEADY-STATE POLLUTANT EMISSIONS DATA: COIL-COOLED PROTOTYPE FURNACE WITH VARIOUS COMBUSTOR LENGTHS AND COOLING FLUIDS

	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
[illegible]	265	1.15	13.3	2.9	45	59	1	0.68	0.970	0.006	3.0	391
	266	0.72	12.6	4.0	30	59	0	0.46	1.038	0.003	2.0	407
	268	1.22	12.0	4.8	21	59	0	0.37	1.067	0.004	1.0	413
	269	1.34	11.4	5.7	30	60	0	0.54	1.157	0.001	0.0	491
	270	1.26	12.0	4.6	21	59	0	0.37	1.068	0.001	0.0	421
[illegible]	271	1.32	11.7	5.4	25	31	1	0.44	0.593	0.010	0.0	260
	272	1.36	11.3	5.9	31	31	1	0.55	0.604	0.010	0.0	252
	273	1.25	12.2	4.5	27	30	0	0.47	0.545	0.005	0.0	271
	274	1.22	12.4	4.0	30	30	0	0.49	0.522	0.004	0.0	266
	275	1.18	13.0	3.3	41	30	0	0.65	0.510	0.002	0.0	263
	276	1.14	13.3	2.7	80	28	2	1.21	0.462	0.018	0.0	259
	277	1.09	13.8	1.8	620	21	54	9.93	0.324	0.444	0.0	259
[illegible]	278	1.10	13.6	2.0	456	21	30	6.64	0.327	0.249	0.0	271
	279	1.23	12.4	4.1	61	25	1	1.01	0.450	0.014	0.0	274
	280	1.19	12.7	3.5	65	26	1	1.02	0.440	0.013	0.0	274
	281	1.15	14.7	3.1	90	23	2	1.36	0.382	0.021	0.0	274
	282	1.32	11.5	5.4	67	23	2	1.20	0.444	0.024	0.0	257
	283	1.29	11.7	5.0	60	21	2	1.03	0.397	0.018	0.0	271

L=0.78m, Water-Cooled	RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
2/17/78-PM	284	1.05	14.3	0.9	198	21	25	2.74	0.311	0.194	1.0	168
	285	1.11	13.4	2.1	20	18	1	0.31	0.298	0.008	0.5	171
	286	1.23	12.2	4.0	18	17	1	0.31	0.312	0.007	0.0	179
	287	1.28	11.8	4.8	17	20	1	0.31	0.374	0.006	0.0	178
	288	1.31	11.5	5.2	17	19	0	0.31	0.373	0.005	0.0	177
	289	1.17	12.7	3.1	16	20	0	0.26	0.339	0.004	0.0	177
2/18/78-AM	290	1.18	13.1	3.4	—*	40	1	—	0.683	0.007	0.0	207
	291	1.12	13.6	2.4	—	40	1	—	0.649	0.007	0.0	204
	292	1.06	14.3	1.3	—	35	2	—	0.535	0.014	0.0	202
	293	1.03	14.8	0.6	—	25	80	—	0.366	0.622	1.0	202
	294	1.23	13.9	4.6	—	21	1	—	0.367	0.006	0.0	207
	295	1.13	13.5	2.5	—	19	2	—	0.319	0.018	0.0	210
	296	1.30	11.8	5.1	—	21	2	—	0.388	0.018	0.0	202
	297	1.12	13.8	2.3	—	19	2	—	0.316	0.020	0.0	207
2/18/78-PM	298	1.13	13.5	2.5	—	19	1	—	0.319	0.013	0.0	204
	299	1.30	11.7	5.0	—	29	0	—	0.406	0.003	0.0	196
	300	1.16	14.7	3.4	—	21	0	—	0.345	0.003	0.0	202
	301	1.10	13.8	2.9	—	19	6	—	0.312	0.050	0.0	199
	302	1.10	13.8	2.0	120	24	3	1.74	0.375	0.025	0.0	188
	303	1.34	11.3	5.6	40	25	1	0.71	0.479	0.010	0.0	207

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*CO meter was inoperative during runs 290 through 301.

Table C-3. STEADY-STATE POLLUTANT EMISSION DATA: OPTIMUM FURNACE WITH FLUE GAS RECIRCULATION

$$\text{Recirc Ratio} = \frac{100 \times (14.49+1.0)}{(SR_{mix}-1.0)14.49} \cdot \left(\frac{\text{kgs of Burned Gas}}{\text{kg of Unburned Air}}\right)$$

RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C	RECIRC RATIO %	T RFG C
304	1.30	11.9	5.1	17	41	1	0.31	0.762	0.006	0.0	293	0.0	
305	1.41	11.0	6.5	120	[illegible]	7	2.25	0.565	0.050	0.0	336	33.5	171
306	1.30	11.9	5.1	60	31	3	1.05	0.573	0.030	0.0	336	38.0	171
307	1.21	12.6	3.8	50	35	2	0.80	0.601	0.022	0.0	329	45.5	171
308	1.11	13.7	2.2	38	45	1	0.57	0.718	0.013	0.0	277	0.0	
309	1.08	14.1	1.6	254	43	30	3.62	0.667	0.244	0.0	274	0.0	
310	1.10	13.8	2.1	130	35	11	1.90	0.563	0.092	0.0	274	56.7	168
311	1.32	14.3	6.7	20	46	1	0.35	0.869	0.006	0.0	279	0.0	
312	1.16	13.1	3.0	15	50	0	0.25	0.826	0.001	0.0	268	0.0	
313	1.36	11.3	5.9	18	54	0	0.34	1.047	0.001	0.5	302	18.8	60
314	1.28	12.0	4.9	20	45	0	0.36	0.831	0.001	0.0	321	31.1	135
315	1.22	12.4	4.0	60	41	0	0.97	0.717	0.003	0.0	329	40.5	177
316	1.11	13.8	2.2	50	52	1	0.73	0.820	0.013		271	0.0	
317	1.22	12.5	4.0	21	52	0	0.36	0.904	0.001	0.0		32.9	52
318	1.09	13.8	1.9	693	40	120	10.02	0.632	0.991	0.0	307	50.3	157
319	1.21	12.6	3.9	70	40		1.13	0.704		0.0	302	32.8	157
320	1.25	12.1	4.4	81	38		1.36	0.682		0.0	327	39.6	171
321	1.21	12.6	3.8	15	50		0.24	0.863		0.0	282	0.0	
322	1.29	11.9	5.0	15	54	0	0.26	1.009	0.001	0.0	291	0.0	
323	1.32	11.6	5.3	16	54	1	0.33	1.029	0.010	1.0	313	21.9	91
324	1.29	11.9	5.0	20	49	1	0.34	0.905	0.010	0.0	321	31.5	152
325	1.29	11.8	5.0	45	44	2	0.77	0.822	0.024	0.0	335	36.7	132
326	1.09	11.9	5.0	80	40	4	1.37	0.750	0.035	0.0	335	37.9	199
327	1.11	13.6	2.1	40	50	90	0.39	0.796	0.752	1.5	271	0.0	
328	1.17	12.8	3.2	20	50	0	0.31	0.844		0.0	277	0.0	
329	1.22	12.4	3.9	18	50	0	0.31	0.879		0.0		0.0	

RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C	RECIRC RATIO %	T RFG C
340	1.20	12.7	3.6	17	51	0	0.28	0.877	0.001	0.0	277	0.0	
341	1.20	12.7	3.6	14	47	0	0.30	0.816	0.003	0.0	316	34.2	135
342	1.19	12.7	3.5	20	45	1	0.31	0.770	0.005	0.0	318	36.1	74
343	1.19	12.8	3.4	35	41	1	0.55	0.695	0.011	0.0	321	44.4	74
344	1.19	12.8	3.5	75	36	4	1.19	0.683	0.040	0.0	324	48.6	74
345	1.19	12.7	3.5	80	35	6	1.26	0.607	0.054	0.0	321	45.8	143
346	1.21	12.4	3.8	30	38	1	0.48	0.661	0.014	0.0	321	39.8	146

Table C-4. STEADY-STATE COMBUSTION GAS COMPOSITION DATA AT TWO LOCATIONS IN THE PROTOTYPE FURNACE

LOCATION C = ENTRANCE TO SECONDARY PORTION OF
THE MAIN HEAT EXCHANGER

LOCATION B = TOP OF CENTRAL CYLINDRICAL DOME

RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
330	1.20	12.7	3.7	10	50	0	0.18	0.867	0.003	0.0	277
C	1.21	12.7	3.8	11	50	0	0.17	0.869	0.003		
331	1.09	13.9	1.9	40	50	1	0.58	0.787	0.008	1.0	271
C	1.06	14.3	1.2	272	51	10	3.81	0.775	0.080		
332	1.14	13.3	2.7	20	51	0	0.30	0.827	0.001	0.0	274
C	1.11	13.7	2.1	60	50	1	0.88	0.795	0.008		
333	1.29	11.9	5.0	11	54	0	0.21	0.999	0.001	0.0	277
C	1.25	12.2	4.5	15	54	0	0.25	0.979	0.001		
334	1.43	10.8	6.7	15	54	0	0.29	1.121	0.001	0.0	291
C	1.37	11.1	6.0	15	54	0	0.27	1.074	0.001		
335	1.16	13.1	3.1	17	51	1	0.28	0.845	0.005	1.0	277
C	1.13	13.4	2.5	40	51	1	0.60	0.827	0.005		
336	1.16	13.1	3.1	18	50	1	0.29	0.837	0.005	0.0	268
B	1.14	13.3	2.6	18	50	1	0.29	0.809	0.005		
337	1.11	13.7	2.1	28	51	1	0.42	0.803	0.007	0.0	271
B	1.07	14.0	1.5	50	49	1	0.71	0.751	0.008		
338	1.07	14.1	1.4	620	50	50	8.76	0.760	0.403	1.5	270
B	1.03	14.5	0.7	711	46	62	9.72	0.681	0.484		
339	1.43	10.8	6.7	15	51	0	0.29	1.046	0.002	0.0	288
B	1.38	11.1	6.1	15	50	0	0.29	0.998	0.002		

Table C-5. STEADY-STATE POLLUTANT EMISSION DATA: PROTOTYPE FURNACE WITH VARIOUS CONFIGURATIONS OF INTERNAL COOLING COILS AND BAFFLES

		RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
SINGLE COIL, L = 0.50 m	WATER	347	1.40	11.0	6.4	20	46	0	0.37	0.932	0.001	0.0	274
		348	1.25	12.2	4.5	20	43	0	0.33	0.779	0.003	0.0	263
		349	1.19	12.7	3.5	20	45	1	0.32	0.771	0.005	0.0	257
		350	1.13	13.4	2.6	50	43	2	0.75	0.702	0.021	0.0	252
		351	1.11	13.5	2.1	130	41	7	1.90	0.648	0.039	1.5	249
		352	1.31	11.7	5.3	17	40	0	0.31	0.763	0.001	0.0	260
	STEAM	353	1.32	11.6	5.4	20	43	0	0.35	0.814	0.001	0.0	271
		354	1.21	12.6	3.8	20	41	0	0.34	0.717	0.003	0.0	263
		355	1.18	13.6	2.4	120	41	4	1.78	0.656	0.038	1.0	256
		356	1.46	10.4	7.0	20	45	0	0.41	0.955	0.001	0.0	279
SINGLE COIL, L = 0.40 m	WATER	357	1.48	10.4	7.2	21	49	0	0.43	1.043	0.001	0.0	266
		358	1.26	12.2	4.6	30	50	0	0.50	0.901	0.003	0.0	249
		359	1.18	13.1	3.4	75	47	3	1.17	0.802	0.024	1.5	241
		360	1.37	11.3	6.0	21	47	1	0.40	0.934	0.006	0.0	239
	STEAM	361	1.43	10.8	6.7	30	40	0	0.57	0.834	0.003	1.5	267
		362	1.33	11.6	5.5	30	48	0	0.53	0.808	0.003	2.5	260
		363	1.24	12.4	4.4	40	45	0	0.66	0.807	0.003	3.0	252
		364	1.26	12.0	4.6	35	45	0	0.39	0.819	0.003	2.0	249
		365	1.43	10.8	6.7	20	46	0	0.40	0.950	0.001	0.0	264
		366	1.29	11.9	5.0	30	47	0	0.51	0.879	0.003	1.0	252
SINGLE COIL + REAR BAFFLE L = 0.50 m	WATER	367	1.38	11.1	6.1	15	46	0	0.28	0.917	0.001	0.0	260
		368	1.30	11.6	5.1	15	46	0	0.26	0.864	0.001	0.0	254
		369	1.23	12.2	4.1	20	46	0	0.33	0.806	0.001	0.0	246
		370	1.17	12.9	3.1	180	45	21	1.85	0.746	0.185	0.0	241
	STEAM	371	1.30	11.7	5.1	20	50	0	0.35	0.932	0.003	0.0	257
		372	1.38	11.0	6.0	20	50	2	0.37	0.998	0.021	0.0	264
		373	1.17	12.7	3.1	27	49	1	0.43	0.818	0.005	0.0	246
DOUBLE COIL + REAR BAFFLE L = 0.50 m	WATER	374	1.34	11.5	5.6	20	33	0	0.37	0.628	0.001	0.0	214
		375	1.19	12.7	3.5	75	31	6	1.18	0.584	0.034	0.0	207
		376	1.15	13.3	2.8	157	31	7	2.40	0.503	0.065	0.0	200
		377	1.29	11.9	4.9	20	38	0	0.34	0.702	0.001	0.0	206
		378	1.23	12.4	4.1	30	33	0	0.49	0.585	0.003	0.0	204
	STEAM	379	1.23	12.3	4.1	30	31	0	0.49	0.540	0.003	0.0	207
		380	1.15	13.3	2.8	120	31	6	1.82	0.512	0.052	1.5	204
		381	1.18	12.9	3.4	55	38	1	0.86	0.527	0.009	1.0	207
		382	1.35	11.3	5.8	20	33	0	0.38	0.638	0.001	0.0	221

NOTE: Optimum burner fired at 1.05 ml/s (1.00 gph)

Table C-5 (continued). STEADY-STATE POLLUTANT EMISSION DATA: PROTOTYPE FURNACE WITH VARIOUS CONFIGURATIONS OF INTERNAL COOLING COILS AND BAFFLES

				RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
DOUBLE COIL + REAR BAFFLE	L=0.40 m	WATER		383	1.37	11.1	6.0	20	39	0	0.38	0.780	0.001	0.0	241
				384	1.30	11.7	5.1	20	40	0	0.36	0.747	0.001	0.0	238
				385	1.23	12.2	4.1	38	38	0	0.64	0.672	0.003	0.0	229
				386	1.19	12.7	3.5	80	35	3	1.26	0.607	0.027	0.0	227
		STEAM		387	1.19	12.7	3.5	55	36	2	0.87	0.624	0.018	0.0	229
				388	1.30	11.7	5.0	30	38	0	0.53	0.717	0.001	0.0	241
				389	1.24	12.3	4.2	35	40	0	0.57	0.717	0.001	1.0	238
				390	1.35	11.3	5.8	28	40	0	0.52	0.788	0.001	0.0	248
	L=0.50 m	WATER		391	1.40	11.0	6.4	30	35	0	0.56	0.721	0.001	0.0	235
				392	1.30	11.7	5.0	20	35	0	0.36	0.645	0.001	0.0	227
				393	1.22	12.4	4.0	35	31	0	0.57	0.548	0.001	0.0	224
				394	1.16	13.1	3.0	100	30	7	1.53	0.502	0.066	0.0	218
				395	1.10	13.6	2.0	≥1600	27	300	≥23.18	0.421	2.484	1.5	211
			[illegible]°C	396	1.16	11.5	2.6	100	33	6	1.53	0.549	0.052	0.0	216
				397	1.21	12.6	3.8	80	31	4	1.28	0.541	0.041	0.0	221
				398	1.27	11.9	4.7	67	31	3	1.15	0.562	0.029	0.0	227
		STEAM		399	1.30	11.7	5.0	20	31	0	0.36	0.582	0.001	1.0	232
				400	1.22	12.4	4.0	35	34	0	0.57	0.599	0.001	0.0	229
				401	1.15	13.1	2.9	115	31	6	1.75	0.508	0.052	0.0	224

Table C-5 (continued). STEADY-STATE POLLUTANT EMISSION DATA: PROTOTYPE FURNACE WITH VARIOUS CONFIGURATIONS OF INTERNAL COOLING COILS AND BAFFLES

			RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
DOUBLE COIL + EXTENDED REAR BAFFLE	WATER	1.0-9.0°-A	402	1.44	10.6	6.7	40	30	0	0.77	0.627	0.001	0.0	216
			403	1.28	11.9	4.9	60	23	4	1.04	0.430	0.035	0.0	207
			404	1.24	12.1	4.3	90	21	6	1.48	0.371	0.057	0.0	204
			405	1.19	12.6	3.5	231	18	37	3.64	0.320	0.333	1.0	199
			406	1.26	12.1	4.5	50	23	1	0.83	0.420	0.014	0.0	214
			407	1.19	12.6	3.5	157	21	15	2.49	0.355	0.135	0.0	211
			408	1.35	11.1	5.6	37	25	0	0.68	0.481	0.003	0.0	227
			409	1.38	11.0	6.0	30	25	0	0.55	0.493	0.001	0.0	232
	STEAM		410	1.38	11.0	6.0	40	23	0	0.73	0.463	0.001	0.0	239
			411	1.26	11.9	4.5	50	20	1	0.64	0.368	0.010	0.0	229
			412	1.21	12.4	3.7	80	21	3	1.28	0.360	0.027	0.0	224
			413	1.14	13.0	2.7	456	19	45	6.90	0.323	0.388	0.0	218
	AIR		414	1.37	11.2	6.0	60	38	7	1.09	0.730	0.078	0.0	243
			415	1.28	11.9	4.9	70	33	5	1.19	0.613	0.049	0.0	243
			416	1.22	12.5	4.0	80	33	4	1.29	0.581	0.048	0.0	241
			417	1.18	12.9	3.3	110	32	4	1.71	0.535	0.040	0.0	239
			418	1.13	13.4	2.5	711	31	42	10.60	0.496	0.357	0.0	237
			419	1.36	11.3	5.9	25	40	0	0.45	0.792	-.000	0.0	254
	NO COIL		420	1.33	11.5	5.5	17	52	0	0.32	0.997	0.001	0.0	268
			421	1.25	12.2	4.4	20	49	0	0.33	0.886	0.003	0.0	261
			422	1.19	12.7	3.6	30	50	1	0.47	0.861	0.005	0.0	254
			423	1.16	13.1	3.0	45	49	1	0.69	0.820	0.013	0.0	252
			424	1.13	13.4	2.4	100	50	5	1.49	0.802	0.043	0.0	249
			425	1.33	11.5	5.5	20	54	0	0.35	1.023	0.005	0.0	277

Table C-5 (concluded). STEADY-STATE POLLUTANT EMISSION DATA: PROTOTYPE FURNACE WITH VARIOUS CONFIGURATIONS OF INTERNAL COOLING COILS AND BAFFLES

		RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
0.05 m DIA COIL+EXTENDED BAFFLE	WATER	426	1.28	11.9	4.9	70	40	2	1.21	0.738	0.018	0.0	238
		427	1.19	12.7	3.5	177	35	15	2.60	0.599	0.135	0.0	227
		428	1.32	11.6	5.3	25	44	1	0.44	0.836	0.010	0.0	241
		429	1.19	12.7	3.5	157	36	11	2.49	0.624	0.099	0.0	227
		430	1.23	12.2	4.1	60	40	2	0.98	0.714	0.019	0.0	231
	STEAM	431	1.23	12.2	4.1	50	40	1	0.82	0.714	0.009	0.0	235
		432	1.31	11.5	5.2	20	44	0	0.35	0.825	0.001	0.0	243
		433	1.21	12.7	3.9	231	36	15	3.71	0.635	0.137	0.0	218
+0.15 m D x 0.81 m L RESONATOR TUBE	WATER	434	1.37	11.3	6.0	20	31	1	0.36	0.605	0.010	0.0	229
		435	1.33	11.6	5.5	15	31	0	0.28	0.587	0.001	0.0	229
		436	1.22	12.7	4.0	111	28	10	1.80	0.493	0.092	0.0	217
		437	1.17	13.1	3.2	426	22	35	6.59	0.365	0.309	0.0	213
		438	1.35	11.5	5.9	17	31	0	0.32	0.600	0.001	0.0	229
	STEAM	439	1.34	11.5	5.7	15	36	0	0.27	0.707	0.001	0.0	235
		440	1.25	12.3	4.5	35	35	1	0.58	0.623	0.009	0.0	227
		441	1.19	12.9	3.6	148	30	9	2.34	0.509	0.081	0.0	221
		442	1.12	13.4	2.5	1259	22	150	18.72	0.359	1.274	1.5	214
		443	1.33	11.6	5.5	15	35	0	0.26	0.680	0.001	0.0	231
DOUBLE COIL+0.35m SPOOL	WATER	444	1.34	11.2	5.6	30	17	0	0.54	0.343	0.001	0.0	174
		445	1.26	12.0	4.5	50	14	0	0.85	0.265	0.001	0.0	174
		446	1.21	12.4	3.9	≥1600	16	3000	≥25.64	0.290	27.471	1.0	166
		447	1.40	11.0	6.4	38	15		0.73	0.318		0.0	185
	STEAM	448	1.35	11.3	5.7	20	43	8	0.36	0.830	0.082	0.0	191
		449	1.25	12.3	4.4	80	35	8	1.32	0.637	0.076	0.0	185
		450	1.19	12.7	3.5	315	23	20	4.97	0.397	0.180	0.0	182
		451	1.27	12.6	5.0	40	31	2	0.68	0.542	0.019	0.0	188

Table C-6. CYCLE-AVERAGED, FLUE GAS POLLUTANT EMISSION CONCENTRATIONS FOR THE MODIFIED, 1.0 ml/s PROTOTYPE FURNACE SYSTEM WITH AN 8-TUBE, LOW-PRESSURE AIR-COOLED COIL, SUPPLEMENTARY HEAT-EXCHANGER INSTALLED

			RUN NO.	STOIC. RATIO	CO2 %	O2 %	CO PPM	NO PPM	UHC PPM	CO GM/KGM	NO GM/KGM	UHC GM/KGM	BACH. SMOKE	TFG C
$\dot{Q}_{A/C} \approx 0.0084$ kg/s	0.0191 m Shroud Gap		491	1.25	12.5	4.5	41	38	7	0.70	0.682	0.066	0.0	222
			492	1.14	13.3	2.6	65	38	5	0.98	0.618	0.047	0.0	219
			493	1.10	13.9	2.0	177	36	22	2.59	0.577	0.183	0.0	217
			494	1.09	13.9	1.9	177	35	25	2.57	0.558	0.207	0.4	221
			495	1.37	11.2	5.9	37	37	12	0.69	0.729	0.125	0.0	229
			496	1.28	12.0	4.9	40	39	8	0.70	0.718	0.078	0.0	235
			497	1.21	12.6	3.8	42	39	6	0.69	0.683	0.057	0.0	229
$\dot{Q}_{A/C} \approx 0.0119$ kg/s	0.0191 m Gap		498	1.21	12.9	3.9	40	38	6	0.64	0.660	0.055	0.4	238
			499	1.06	13.5	1.2	-1600	68	3000	-22.36	1.024	23.960	5.0	210
			500	1.14	13.3	2.8	67	41	30	1.03	0.670	0.259	0.0	227
			501	1.13	13.7	2.6	70	37	15	1.05	0.608	0.128	0.0	237
			502	1.36	11.4	6.0	30	39	9	0.54	0.775	0.093	0.0	257
	0.0381 m Shroud Gap	Rear Baffle	503	1.24	12.4	4.4	40	40	8	0.66	0.722	0.075	0.0	213
			504	1.05	14.2	1.1	620	31	40	8.64	0.463	0.318	0.0	207
			505	1.18	13.0	3.4	50	40	6	0.78	0.684	0.054	0.0	210
			506	1.12	13.6	2.4	87	35	10	1.30	0.572	0.085	0.0	207
			507	1.32	11.8	5.4	37	38	9	0.66	0.720	0.090	0.0	218
			508	1.29	11.9	5.0	30	43	6	0.51	0.794	0.059	0.5	241
			509	1.22	12.5	3.9	35	44	4.	0.56	0.771	0.041	0.2	231
			510	1.38	11.0	6.1	35	40	7	0.64	0.806	0.074	0.0	245
			511	1.42	10.8	6.6	27	39	8	0.53	0.781	0.087	0.0	249